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DIMENSIONAL ANALYSIS OF MODEL PROPELLER TESTS.*

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PREFACE.

The following paper does not profess to contain anything that is not already perfectly familiar to model-basin experts and it is not intended for them. But I suspect that there may be some members of the American Society of Naval Engineers who, like myself, have never had an opportunity of actual contact with the experimental side of the subject and yet are interested in the principles involved in the conduct and interpretation of experiments on model screw propellers. I hope that this paper, which has been written mainly for my own instruction, may be of some use to such readers, as an introduction to the subject.

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Since it makes no pretense to completeness, I have not thought it necessary to give many references, but a few will be found in the bibliography and many more may be found by consulting the papers there listed.

I must especially acknowledge the advantage I have had, while writing the paper, from numerous conferences with Commander Lybrand P. Smith, U.S.N., D.Sc., and from access to the results of his experiments on model propellers.

I. INTRODUCTION.

When a machine or other structure is to be built for a particular purpose, the desired performance or behavior is specified, and the problem before the designer is, to get this result in the most advantageous—usually the most economical—way. If enough experience with similar structures and requirements is available, the solution may be fairly definite; but in the absence of such experience there is often a good deal of uncertainty, and after the structure has been designed and built it may be found to need expensive alterations. In many such instances, this might have been foreseen and avoided by making relatively inexpensive preliminary tests on models representing various tentative designs, before making the final decision and proceeding to build.

The structures to which this paper refers are ships driven by screw propellers, and the problem to be studied by means of experiments on models is usually somewhat as follows: The displacement of a proposed ship and the required speed are given, and the general shape of the under-water body is approximately determined by the requirements of service in connection with past experience in hull design: the problem is, to find a combination of hull and propeller, or propellers, that will give the required speed for the least power.

The value of model experiments depends on the possibility of predicting the behavior of a full-sized ship from the observed behavior of the model, and the fundamental condition for this is geometrical similarity of the immersed parts. If a tentative design for a ship is under consideration, the model must represent this design, in all details of external shape that have any sensible influence on the behavior of the model. This means, in strictness,

that the model hull and propeller should be tested as a unit; for each is affected by the presence of the other and they will have to run together in the completed ship. But although these mutual influences are important, useful information may nevertheless be obtained by testing the model hull and model propeller separately and employing the more complicated tests of the combination only for determining corrections to be applied to the results of the separated tests.

Assuming that the matter of shape has been attended to, the working conditions must be considered. In order that the behavior of the proposed ship when running at a given speed in smooth water may be reliably predicted from observations on the model, the circumstances of the model experiment must be so arranged and adjusted that the test is " dynamically similar " to the full scale run which it purports to imitate; and *the purpose of this paper is to deduce and discuss the conditions for dynamical similarity.* They may be obtained by the detailed, step-by-step method used by Newton; but the modern " method of dimensions," due to Rayleigh and also known as " dimensional analysis," is simpler and will be followed here.

The presence of surface waves from other sources than the motion of the model itself complicates matters almost indefinitely; and since only a few artificially simple cases can be imitated in the model basin or towing tank, the discussion will be limited to the consideration of runs in smooth water.

We shall be concerned primarily with propellers and it will at first be supposed that the propellers are working normally, without cavitating. The discussion will then be extended to the somewhat more complicated phenomena of cavitation, which have not yet been so thoroughly investigated by experiment and are therefore of more present interest.

II. PROPELLER WORKING WITHOUT CAVITATION.

1. *The Independent Variables.* The various effects which are to be studied quantitatively by means of experiments on models—thrust and torque of a propeller, tow-rope resistance of a bare hull, etc.—are due to the fact that water of a particular density and viscosity is moving in a certain manner relatively to the model and

is exerting normal pressures and tangential stresses on its surface. The space distribution of the speed and direction of the motion of the water about the model, or the velocity field, is too complicated to be described or specified in detail. But it is possible to make up a rather short list of measurable quantities, such that when their values are fixed the velocity field about the model is thereby determined,—not exactly but nearly enough for practical purposes. And after such a set of quantities has been found, the dependent quantities, in which we are more directly interested, may be treated as functions of them. Our ultimate object is to find out as much as possible about the forms of these functions, but we begin by fixing our attention on the motion of the water and we ask ourselves: What are the essential characteristics of a model experiment which determine that the relative motion of the water and its reactions on the model shall be what they are instead of something different?.

Starting with a particular model test, we may change the forward speed or the rate of rotation of the propeller or both, and any such change in the working conditions will produce some definite alteration in the motion of the water relative to the model. We next consider the density and viscosity of the water. These can be varied only by changing the temperature, and no great variations are practicable. But much greater changes might conceivably be brought about by substituting other liquids for water; and the fact that the density and viscosity do have particular, nearly constant, values is evidently an essential and determining characteristic of the flow.

In general, a moving hull throws up a wave system, or a propeller disturbs the free water surface above it, unless deeply submerged. The magnitude and shape of these disturbances depend on the specific weight of the water, which is proportional to the product of its density and the intensity of gravity. Hence if the value of g were different from what it is, the motion of the water would also be different: the fact that g has a particular value is an essential characteristic of the phenomena and g has to appear in our list.

The decisive importance of density, viscosity, and weight may be appreciated if one tries to imagine how the behavior of a model might be affected if the tests could be run in lubricating oil or mercury instead of in water.

We come finally to the question of size. The shape of the model is fixed by that of its full-sized counterpart, but we can use larger or smaller models and so change the size independently of the values of the other quantities mentioned above. Any such change will evidently alter the velocity field and the forces on the model; so the size must be specified, and the simplest way to do it is to state the value of some linear dimension—any one will do. In dealing with a hull, the load-water-line length is the most natural dimension to use, but any other characteristic length, such as the square root of the wetted area or the cube root of the displaced volume, would serve the same purpose. Being interested mainly in propellers, we shall generally take the propeller diameter as the characteristic dimension.

We now have the following list of quantities, with their dimensions in terms of mass, length, and time:

Symbol	*Dimensions*	
D = diameter of the propeller	l	(I)
n = number of revolutions per unit time	t^{-1}	
v = forward speed of the model	lt^{-1}	
ρ = density of the water	ml^{-3}	
μ = viscosity of the water	$ml^{-1}t^{-1}$	
g = acceleration of gravity	lt^{-2}	

With a model of fixed shape and so long as there is no cavitation, the values of these six quantities determine, within narrow limits, how the water shall move with respect to the model and what forces it shall exert on the model. The determination is not exact because we have taken no account of various things, such as the density of the air above the water surface; but the remaining uncertainties in the forces on the model are insignificant in comparison with the errors of measurement that are unavoidable in model experiments, to say nothing of trial runs of ships at sea. For our present purposes, no other quantities than those listed above are of any practical importance, but all six must be included in any general discussion of ship-model experiments, although in particular cases some of them may be disregarded and the list abbreviated.

The list may, however, be modified in appearance and still serve the same purpose; for any other set of six quantities whose values

determine the values of those already considered would be mathematically equivalent to them; and beside the occasional substitution of some other length for D, two other changes are advisable. The first consists in replacing the viscosity μ by the kinematic viscosity $\nu = \mu/\rho$, which fixes μ when ρ is also given. This substitution simplifies the equations with which we shall have to deal. Another convenient substitution is that of the slip ratio S for the rate of rotation n. Since the given shape of the propeller determines its pitch ratio a, we have the relation

$$S = 1 - \frac{v}{D\,a\,n}$$

so that when D and v are given, S determines n and vice versa.

With these modifications we have the list:

Symbol	*Dimensions*	
D = diameter of the propeller	l	(II)
v = forward speed	lt^{-1}	
S = slip ratio	1	
ρ = density of the water	ml^{-3}	
ν = its kinematic viscosity	l^2t^{-1}	
g = acceleration of gravity	lt^{-2}	

For the purpose of determining the velocity field and its effect on the model, this list is entirely equivalent to the previous one. On account of the role they are to play, these quantities will, for convenience, be referred to as the "independent variables," in spite of the fact that only D, v, and S (or n) can be varied at will, while ρ and ν (or μ) are nearly constant in practice and g is altogether constant at any one place.

2. *Formal Statement of the Conditions for Dynamical Similarity.* (1, 2, 3)*

From a list of 6 quantities with dimensions stated in terms of 3 fundamental quantities, it is possible to make up $6 - 3 = 3$ expressions of the general form

$$\Pi = Q_1^x\, Q_2^y\, Q_3^z\, P \qquad (1)$$

where the Q's are 3 of the quantities and P is each of the remain-

* Throughout the text, numbers in parentheses will refer to the bibliography at the end except where they obviously refer to numbered equations or expressions.

ing 3, in succession; and to determine the 3 sets of values of (x, y, z) in such ways that each of the 3 Π's shall be "dimensionless," or of zero dimensions with respect to each of the 3 fundamental quantities.

In the second of the lists given above, the slip ratio S is itself dimensionless, as might be indicated by writing its dimensional equation in the form $[S] = [m^0\, l^0\, t^0]$ instead of in the usual more compact form $[S] = [1]$. Disregarding S for the present, we proceed to form 2 dimensionless combinations of the remaining 5 quantities, 4 at a time. This might be done in 6 different ways but for our present purpose it is expedient to let $Q_1, Q_2, Q_3 = D, \rho, v$, and

$$\left.\begin{aligned} \Pi_\nu &= D^x\, \rho^y\, v^z\, \nu \qquad (a) \\ \Pi_g &= D^a\, \rho^b\, v^c\, g \qquad (b) \end{aligned}\right\} (2)$$

Substituting in the first of these equations the dimensions of D, ρ, v, and ν as given in the list, we find that the dimensions of Π_ν are

$$[\Pi_\nu] = [l^x \cdot m^y\, l^{-3y} \cdot l^z\ \ t^{-z} \cdot l^2\, t^{-1}]$$
$$= [l^{x-3y+z+2} \cdot m^y \cdot t^{-z-1}]$$

To make the dimensions vanish, we must let

$$\left.\begin{aligned} x - 3y + z + 2 &= 0 \\ y &= 0 \\ -z - 1 &= 0 \end{aligned}\right\} \text{ or } \left\{\begin{aligned} x &= -1 \\ y &= 0 \\ z &= -1 \end{aligned}\right.$$

so that, by equation (2, a), $\Pi_\nu = \nu/Dv$. A similar procedure gives the values that (a, b, c) must have in order to make the dimensions of Π_g vanish, and we find that the expression $\Pi_g = Dg/v^2$ is dimensionless. Returning to S for the third possible expression of the form indicated by equation (1), we might, as a matter of form, treat S in the same way as ν and g, by writing

$$[\Pi_s] = [D^\alpha\, \rho^\beta\, v^\gamma\, S] = [l^\alpha \cdot m^\beta\, l^{-3\beta} \cdot l^\gamma\, t^{-\gamma} \cdot m^0\, l^0\, t^0]$$

which would give us $\alpha = \beta = \gamma = 0$, and $\Pi_s = S$.

From the 6 independent variables we have now made up the 3 dimensionless combinations:

$$\left.\begin{aligned} \Pi_s &= S && \text{(a)} \\ \Pi_\nu &= \frac{\nu}{Dv} && \text{(b)} \\ \Pi_g &= \frac{Dg}{v^2} && \text{(c)} \end{aligned}\right\} (3)$$

and *the conditions for dynamical similarity* between a model test and a test run of the full-sized ship, or between tests of any two geometrically similar models, are that *each of the 3* Π*'s must have the same numerical value in both tests.*

To be more specific, let subscript 0 refer to the full-scale run and subscript 1 refer to the model test.

The diameter D_0 of the large propeller is given; the speed v_0 and the slip ratio S_0 are specified; the value of the kinematic viscosity ν_0 is determined by the temperature of the water in which the ship is to run; and the value of g_0 is known for any given geographical location. The value of D_1 is fixed by the scale ratio to which the model has been constructed, and the value of g_1 is known and fixed. The value of ν_1 depends on the temperature, which usually can not be varied arbitrarily or, at most, only over a short range; but the forward speed v_1 and the slip ratio S_1 are under control. For dynamical similarity it is therefore necessary to adjust the values of v_1 and S_1 in the model test so as to satisfy the equations

$$\left.\begin{aligned} S_1 &= S_0 && \text{(a)} \\ \frac{\nu_1}{D_1 v_1} &= \frac{\nu_0}{D_0 v_0} && \text{(b)} \\ \frac{D_1 g_1}{{v_1}^2} &= \frac{D_0 g_0}{{v_0}^2} && \text{(c)} \end{aligned}\right\} (4)$$

Since the variations of gravity at sea level between the latitudes ± 70 degrees are less than 0.5 per cent, it is permissible to set $g_1 = g_0$; and if we denote the scale ratio by $D_1/D_0 = r$, equations (4) may be put into the equivalent forms

$$\left.\begin{aligned} S_1 &= S_0 && \text{(a)} \\ v_1 &= \frac{v_0}{r} \cdot \frac{\nu_1}{\nu_0} && \text{(b)} \\ v_1 &= v_0 \sqrt{r} && \text{(c)} \end{aligned}\right\} (5)$$

The expression ν/Dv is the reciprocal of the familiar "Reynolds number," $R = Dv/\nu$, so that equation (4, b) is equivalent to $R_1 = R_0$. And if we abbreviate farther by letting $Dg/v^2 = F$, equations (4) may be written more concisely in the forms

$$\left.\begin{aligned} S_1 &= S_0 \quad (a) \\ R_1 &= R_0 \quad (b) \\ F_1 &= F_0 \quad (c) \end{aligned}\right\} (6)$$

In whichever of these equivalent forms, (4), (5) or (6), it may be expedient to express the conditions for dynamical similarity, the first may be referred to as the "slip condition," the second as the "Reynolds condition," and the third as the "Froude condition."

Although the first and third conditions present no serious difficulty, it is nearly always impracticable to satisfy the second, so that complete dynamical similarity is unattainable and we have to be content with more or less close approximations to it. But before discussing this point in detail we may proceed to show why the conditions, of which formal statements have just been given, must be at least approximately satisfied if model experiments are to furnish a reliable guide for predictions regarding full-scale performance.

3. *The Importance of Dynamical Similarity.* Let T be the thrust of a propeller; its dimensions are $[T] = [m\,l\,t^{-2}]$. The value of T is determined by the values of the six independent variables; in other words, there is some definite relation which may be represented symbolically by an equation

$$T = f\ (D, v, S, \rho, \nu, g)$$

or

$$f_1\ (T, D, v, S, \rho, \nu, g) = 0 \qquad (7)$$

The detailed form of the equation will depend on the shape of the propeller and of the hull behind which it works, and we have no means of knowing *a priori* what the form may be: if we had, we could compute the thrust of the full-sized propeller without resorting to model experiments. But we know that some such equation must exist, and furthermore that whatever its form may be and however many terms it may have, all the terms must be of the same dimensions if the equation is to be generally correct.

In order that this may certainly be true for any and all forms of the unknown operator or functional symbol f_1, it is necessary and sufficient that all the operands be separately of zero dimensions. Consequently, equation (7) must be reducible to the form

$$f_2\ (\Pi_1,\ \Pi_2, \text{- - - etc.}) = 0 \tag{8}$$

in which the Π's are dimensionless combinations of the 7 variables which appear in equation (7) but, with the exception of S, are not dimensionless as they stand.

The simplest way to make up such combinations is to form dimensionless products of powers of the variables as was done in the preceding section, letting each Π contain some one variable that does not appear in any of the others, so that the Π's are all independent, in the sense that it is impossible to compute the value of any one from the values of the others. The number of such independent combinations is evidently limited by the number of variables, and after as many as possible have been found they are to be inserted in equation (8). We might go on indefinitely making up further dimensionless combinations, but since their values would be fixed by the values of those already found there would be no object in including them in equation (8); for so long as the form of f_2 remains undetermined, the meaning of the equation would not be affected thereby.

With 7 quantities and 3 fundamental units, the number of independent Π's is $7 - 3 = 4$. Three have already been found and are given in equations (3). The fourth is to be found by setting

$$\Pi_T = D^x \rho^y v^z T$$

substituting the dimensions, and determining (x, y, z) as before. The result is

$$\Pi_T = \frac{T}{D^2 \rho v^2} \tag{9}$$

and we conclude that the relation connecting T with the 6 independent variables must be expressible in the form

$$f_2 \left(\frac{T}{D^2 \rho v^2}, S, R, F \right) = 0 \tag{10}$$

or, after solving for Π_T and multiplying by $D^2\rho v^2$,

$$T = D^2\rho v^2 \varphi\ (S, R, F) \qquad (11)$$

Since the form of φ remains unknown, this equation does not, by itself, convey any definite information; but let us write

$$\left.\begin{aligned} T_0 &= D_0^2\rho_0 v_0^2 \varphi\ (S_0, R_0, F_0) \\ T_1 &= D_1^2\rho_1 v_1^2 \varphi\ (S_1, R_1, F_1) \end{aligned}\right\} \qquad (12)$$

which are the forms assumed by equation (11) for the full-scale and model tests, respectively. Then if equations (6) are satisfied, the undetermined parts of equations (12) become identical, regardless of the form of φ; and by division we get the *definite* statement that

$$\frac{T_0}{T_1} = \left(\frac{D_0}{D_1}\right)^2 \frac{\rho_0}{\rho_1}\left(\frac{v_0}{v_1}\right)^2 \qquad (13)$$

If we start with the torque Q instead of with the thrust; set $\Pi_Q = D^x \rho^y v^z Q$; insert the dimensions $[Q] = [ml^2t^{-2}]$; and determine (x, y, z) so as to make Π_Q dimensionless; we find that

$$\Pi_Q = \frac{Q}{D^3\rho v^2}$$

Hence, proceeding as before, we obtain the equation

$$Q = D^3\rho v^2 \psi\ (S, R, F) \qquad (14)$$

which, like equation (11), contains an undetermined function of (S, R, F) and does not tell us anything definite. But if the circumstances of the model test are so arranged that equations (6) are satisfied, we can get rid of the unknown function and obtain the definite equation

$$\frac{Q_0}{Q_1} = \left(\frac{D_0}{D_1}\right)^3 \frac{\rho_0}{\rho_1}\left(\frac{v_0}{v_1}\right)^2 \qquad (15)$$

To take the simplest type of quantity, let α be the angle through which the fore-and-aft trim of the hull is altered when running at the speed v. An angle is a pure ratio and has no dimensions, so we set $[\alpha] = [1]$ and by proceeding as before we find that $\Pi_\alpha = \alpha$ and arrive at the indefinite equation

$$\alpha = \psi_1 (S, R, F) \tag{16}$$

In this case it is natural to specify the size by the length L of the hull rather than by the propeller diameter, and if we do so, the Reynolds number will be $R = Lv/\nu$.

If equations (6) are satisfied, equation (16) leads to the definite statement that

$$\frac{\alpha_0}{\alpha_1} = 1 \qquad \text{or,} \qquad \alpha_0 = \alpha_1 \tag{17}$$

and if the model test is dynamically similar to the full-scale run, the angle of change of trim will be the same in both.

With any other dimensionless ratio, such as the wake fraction or the thrust-deduction coefficient, the result will be equally simple; and if the model test meets the requirements of dynamical similarity, the value determined with the model may be applied directly to the full-sized ship, without any intermediate computation.

More generally, with any quantity that is determined by the forces due to the motion of the water past the hull and propeller and may therefore be treated as a function of the independent variables listed in section 1, the procedure illustrated above always leads to an equation analogous to (11), (14), or (16), which is true for both the ship and its geometrically similar model but does not tell us anything definite about either of them or about their relative behavior. But this difficulty may be circumvented if the model test can be so arranged as to satisfy the conditions of dynamical similarity formulated in equations (4), (5), and (6), and we then get a definite statement which makes it possible to predict full-scale performance from observations on a small model.

4. *Discussion of the Reynolds Condition.* We return to equation (5, b) or

$$v_1 = \frac{v_0}{r}\frac{\nu_1}{\nu_0}$$

Let us suppose, for simplicity, that the ship and the model are to run in the same kind of water and at the same temperature: then $\nu_1 = \nu_0$ and the Reynolds condition requires that $v_1 = v_0/r$. But in order to make model experiments practicable and worth while, the scale-ratio, $r = D_1/D_0$, must be a small fraction; so the Reynolds condition demands that the model be run many times faster than the designed speed of the ship,—which is quite impossible with any existing equipment. If the ship were to run in water at 40 degrees F. and the water for the model test could be heated to 100 degrees, the value of ν_1/ν_0 could be reduced to about 0.45, but the speed required for the model would still be impossibly high.

In view of this difficulty, it is hardly necessary to mention the impossibility of satisfying equations (5, b) and (5, c) simultaneously. That would require that

$$\frac{1}{r}\frac{\nu_1}{\nu_0} = \sqrt{r} \qquad \text{or} \qquad \nu_1 = \nu_0 r^{\frac{3}{2}}$$

and with r a small fraction, any such lowering of the kinematic viscosity of the water is out of the question.

Since it is thus impossible to satisfy the Reynolds condition, we have to consider the probable effects of this unavoidable departure from complete similarity on the accuracy of predictions based on the results of experiments on models.

The presence of the Reynolds condition among the requirements for dynamical similarity was due to the inclusion of ν in the list of independent variables. If it were certain that variations of the viscosity had no appreciable influence on the forces exerted by the water on the hull and propeller, ν might be omitted from the list and the Reynolds condition would disappear. We must therefore consider the question: In what cases is viscosity likely to be of serious importance; and when, if ever, may it safely be disregarded?.

In any fluid motion that is not strictly uniform, contiguous filaments are moving at different speeds and their interaction tends to retard the faster and accelerate the slower. In quiet, stream-line flow this tangential drag is transmitted by viscosity; but in turbulent, eddying flow there is a transverse interchange of momentum by

cross mixing, which has the same effect of equalizing the speeds as if the viscosity had been increased with the turbulence absent. This effect is known as the "mechanical" or "turbulent" viscosity, in distinction from the ordinary "physical" viscosity given in the physical tables, which is a specific property of the fluid and does not depend on its state of motion.

In regions of active turbulence, even though the individual eddies be small and the turbulence fine-grained, the mixing may have so much more effect than the true viscosity that variations in the latter have very little influence on the configuration of the flow. But where a fluid is moving along a solid surface there is always a very thin layer into which no turbulence can penetrate from farther out, so that the tangential drag between the main stream and the solid has ultimately to be transmitted through this thin stream-line layer by viscosity. Hence in any phenomenon that is much influenced by skin friction, the value of the viscosity may be a matter of considerable importance.

The forces on the blades of a propeller that is exerting a thrust are due mainly to normal pressures, and the tangential stresses of skin friction are not of much importance so long as the blades are smooth. If the blades are rough, skin friction has serious effects on both thrust and torque; but it will be assumed, throughout, that the propellers under discussion are smooth,—as they must be in order to perform satisfactorily or even to permit of accurate model tests. For although tests of roughened models may furnish qualitative information in the shape of warnings against the bad effects of roughness, it is impossible to specify irregular roughness such as may occur on the blades of a large propeller after it has been in service, or to reproduce it to scale on a model.

When a propeller is running in the wake of a hull the stream that meets it is already turbulent, whereas if the propeller is advancing into quiet water there is no turbulence except what is caused by the propeller itself and trails away aft. It may therefore be anticipated that if variations of viscosity are ever to cause appreciable variations in thrust and torque, it will be in open-water tests, where turbulence is least likely to mask the effects of true viscosity; and the effects in question are, in fact, sometimes noticeable in open-water tests of small model propellers. Some further

remarks on this subject will be given later under the heading " Scale Effect;" but it will be sufficient here to state that the effects of viscosity, which manifest themselves in a dependence of thrust and torque on the Reynolds number, appear to be of minor importance in comparison with other difficulties and uncertainties in interpreting the results of open-water tests.

With the hull, matters are somewhat different. For while the tangential forces of skin friction on the blades of a smooth propeller are relatively insignificant, the forces on the hull and the character of the wake are very largely dependent on skin friction. Separated tests of model hulls and model propellers may therefore be discussed somewhat differently, and a few remarks on hull testing may be interjected here before proceeding with our main subject of propellers.

5. *Experiments on a Model Hull.*

(a) *Wake.* When a hull of length L is towed at the speed v, with the propeller absent, the motion of the water about it is determined by the values of L, v, ρ, ν, and g. At any given point in the wake, the angle between the direction of flow and the axis of the propeller shaft depends directly on the values of R and F, in accordance with some equation of the same general form as equation (16) but with S omitted and D replaced by L in the expressions for R and F. The ratio of the speed of the water at the point in question to the forward speed of the hull is also dimensionless and is related to R and F by some analogous equation, in which L does not appear explicitly.

Since this is true for every point in the wake, the distribution of speed and direction throughout the whole wake is determined by the values of R and F; and the wake of a model hull will be in all respects similar to that of the full-sized ship when towed without its propeller, if the Reynolds and Froude conditions are satisfied. And if w is the ratio of the average forward speed of the water at the propeller circle to the forward speed of the hull, there will be some relation

$$w = \varphi\ (R, F) \qquad (18)$$

which does not involve L explicitly and is therefore the same for any two hulls of the same shape, regardless of size.

When the propeller is absent, it is possible to make a complete Pitot tube survey of the wake of a model hull, or to determine its average effective speed by means of fan-type current meters. Such investigations may furnish valuable suggestions for improving the lines of the after-body, the shape and position of struts or bossing, or the location of propellers. But when a hull is being driven by its own propeller, the wake is influenced by the action of the propeller, and it can not be assumed that a value of w found from experiments on the model hull alone will be immediately applicable to the complete, full-sized ship. To eliminate this uncertainty, experiments must be made with the propeller running in its designed position behind the hull, and we then have the more general statement that

$$w = \varphi\ (S,\ R,\ F) \tag{19}$$

If skin friction played only a negligible part in producing the wake, the model and full-sized wakes could be made similar by satisfying the slip and Froude conditions. But since the important frictional element of the wake depends on viscosity and therefore on the Reynolds number, running at equal slip ratios and at corresponding speeds is not enough. And since it is impossible to satisfy the Reynolds condition, a value of w determined for the model, by whatever method, may be considerably in error for the full sized ship at the same S and F. It is likely to be too high, especially for single screw ships.

(b) *Tow Rope Resistance.* If X is the resistance, the initial equation corresponding to equation (7) is

$$f\ (X,\ L,\ v,\ \rho,\ \nu,\ g) = 0$$

And since X has the same dimensions as thrust, we obtain an equation of the same form as (11) with S omitted, or

$$X = L^2 \rho v^2 \varphi\ (R,\ F) \tag{20}$$

It is now impossible to attain a sufficient approximation to similarity without satisfying the Reynolds condition, because skin friction, which depends largely on viscosity, is always a major element

in the resistance. It is therefore necessary to resort to the simplifying assumption that the resistance consists of two separate and independent parts, one due to skin friction and the other, the so-called residuary resistance, due to the continual generation of waves and eddies. This is evidently only an approximation, but it represents the facts well enough to be very useful and we may express it by writing

$$X = V + W \tag{21}$$

where V is the frictional resistance, which depends on viscosity but not on gravity; and W is the residuary resistance, due mainly to wave formation, which depends on gravity but not on viscosity.

The assumption amounts to setting

$$\left.\begin{aligned} V &= f_1 (L, \rho, v, \nu) \\ W &= f_2 (L, \rho, v, g) \end{aligned}\right\} \tag{22}$$

whence the usual procedure leads to the equations

$$\left.\begin{aligned} V &= L^2 \rho v^2 \varphi (R) \qquad \text{(a)} \\ W &= L^2 \rho v^2 \psi (F) \qquad \text{(b)} \end{aligned}\right\} \tag{23}$$

Let the ship be run at the speed v_0 and the model at the "corresponding" speed, $v_1 = v_0 \sqrt{L_1/L_0} = v_0 \sqrt{r}$. Then $F_1 = F_0$ and equation (23, b) gives us the relation

$$\frac{W_0}{W_1} = \frac{L_0^2 \rho_0 v_0^2}{L_1^2 \rho_1 v_1^2} \tag{24}$$

whence

$$W_0 = W_1 \frac{1}{r^3} \cdot \frac{\rho_0}{\rho_1} \tag{25}$$

But W_1, the residuary resistance of the model, can not be measured; for, as the name implies, it is merely what is left over after subtracting the frictional resistance of the model from the observed

total resistance; and it is equally impossible to separate out the frictional resistance and measure it by itself. What has therefore to be done is to compute the frictional resistance of the model from its wetted area and the data obtained from friction plank experiments, and set $W_1 = X_1 - V_1$. This value, substituted in equation (25), gives W_0, the residuary resistance of the full sized hull. The frictional resistance V_0 is then computed from the wetted area of the large hull, in the same way as V_1 for the model; and the total resistance of the full sized ship is then taken to be

$$X_0 = V_0 + W_0$$

The great source of uncertainty in the final result of this roundabout process is, of course, in the computation of the frictional resistances. But since this paper is concerned primarily with screw propellers, no attempt will be made here to discuss the very large subject of skin friction or the various methods that have been employed for analyzing, interpreting, and utilizing the great mass of existing experimental data. For further information the reader may consult recent papers by Schoenherr (4), Hiraga (5), and Bairstow (6), with the accompanying discussions and further references.

6. *Open-Water Tests of a Model Propeller.*

(a) *Wake fraction.* Before the results of an open-water test of a propeller can be used for predicting its performance behind a hull, allowance must be made for the difference in behavior of the same propeller—necessarily a model—when advancing through quiet water and when running in the turbulent and nonuniform wake of the hull. This is done by means of a "wake fraction," which may be regarded as representing, rather roughly, the ratio of the average forward speed of the wake to the forward speed of the hull.

As noted in section 5, a, some information about the wake of a model hull may be obtained from experiments on the hull alone; but the only convincing method of determining a wake fraction is to compare tests of the propeller, with and without the hull in front of it. This is done as follows, or by some equivalent procedure:

Let the propeller be in its designed location behind the hull but supported independently from farther aft, so that the hull can be removed without disturbing the propeller. With the combination moving at the speed v and the propeller turning at the rate n, let the observed thrust be T. Then let the hull be removed and, with n unchanged, let the speed of advance be adjusted so that the thrust has its former value T. If v_a is this new speed, the wake fraction w is defined by setting

$$w = \frac{v - v_a}{v} \quad \text{or} \quad v_a = (1 - w)\, v \qquad (26)$$

Conversely, if the value of w is already known, the thrust T, which the propeller would exert, at the rate of rotation n, behind the hull at the forward speed v, may be found by measuring the thrust in an open-water test, at the given n and at the speed v_a given by equation (26).

If every "open" test had to be supplemented by a "behind" test before its results could be utilized, the open tests would evidently be superfluous. But the value of w is not very sensitive to small changes in the shape of the hull, and is still less so to changes in the shape of the propeller; and it may usually be inferred, with fair accuracy, from the known results of previous experiments on approximately similar combinations of hull and propeller. Thus, by assuming a value of w, the effects of varying the size and shape of the propeller on its performance behind its hull may be found, approximately, from open tests alone. And after a design which appears satisfactory has been found, the uncertainty involved in the use of an assumed value of w may be removed by a final test of the combination.

Changing the speed from v in the behind test to v_a in the open test is equivalent to replacing the wake in which the propeller works when behind the hull by a quiet, straight, uniform current moving forward at the speed wv. The manner in which the propeller meets the water is very different in the two cases, and although v_a has been adjusted so that the thrusts are equal in the open and behind tests, the torques may not be equal. A wake fraction might be defined for equal torques instead of equal thrusts, and the two

values might differ. But the actually observed differences are small, and in practice w is defined and determined with respect to thrust. The torque to be expected behind the hull may then be computed from that observed in the open test by the application of a correction factor, which does not differ much from unity and may be either determined directly or inferred, with sufficient accuracy, from previous experience.

(b) *Depth of Submergence.* When a propeller is running near the surface, its suction depresses the surface above it and its thrust and torque are somewhat dependent on the amount of depression and on the specific weight of the water. The effect is insignificant unless the blade tips come rather close to the surface, and if the propeller is so deeply submerged that lowering it farther has no appreciable effect on its thrust and torque, the disturbance of the water surface and the weight of the water are no longer of any importance in determining the performance of the propeller. Under these conditions, g may be omitted from the list of variables, when treating the propeller apart from the hull, and no further attention need be paid to the Froude condition.

Experiments on model propellers of various shapes and sizes, by D. W. Taylor (7) and by G. S. Baker and W. C. S. Wigley (8), have shown that at ordinary slip ratios ($S \gtrless 0.3$) decreasing the submergence has no serious effect on thrust and torque until the depth h, to the center of the hub, is $0.7D$ or less. And experiments by G. Kempf (9) have confirmed this result in more detail, for three propellers of a single shape but different diameters.

Propellers for sea-going vessels are usually designed for a normal submergence considerably greater than $h/D = 0.7$. An examination of data quoted by L. P. Smith (10) for 63 sea-going vessels of various types and sizes, selected at random from naval and mercantile examples, shows only 6 cases in which h/D was less than 0.7. And only one of these, an early aircraft carrier, was an important naval ship, the others being a repair ship, an ocean tug, and three freight or passenger ships.

It may be concluded that in nearly all wake-fraction determinations, after the self-propelled or behind test has been carried out, the open test may be made at a greater depth of submergence, if

this happens to be more convenient, and that this will not affect the value of the wake fraction.

In exceptional cases, where the designed submergence is less than $h/D = 0.7$, if the open test is made at a sufficient depth to eliminate the effect of surface disturbances, the wake fraction may be slightly different from the value it would have if the open test were made at the same depth as the behind test. But since $(1 - w)$ is merely a correction factor for reducing the thrust in an open-water test to the value it would have in a self-propelled test, the submergence in the open test is, in reality, a matter of no importance, so long at it is such as always to give the same value of w. This will be true if the submergence in the open test is made sufficient to reduce the effects of surface disturbances to insignificance; and when this is done, the Froude condition for dynamical similarity between the model and full sized propellers need not be satisfied.

(c) *Thrust and Torque Coefficients.* We suppose that all open tests are made at so great a depth that the weight of the water and the value of F may be disregarded. The thrust and torque equations, (11) and (14) of section 3, then take the forms

$$\left.\begin{aligned} T &= D^2 \rho v^2 \varphi_1 (S, R) \quad \text{(a)} \\ Q &= D^3 \rho v^2 \psi_1 (S, R) \quad \text{(b)} \end{aligned}\right\} (27)$$

The speed v is here to be understood as the speed of advance *in open water,* previously denoted by v_a, so that T is the thrust which the propeller would exert if running behind its hull at the speed $v/(1 - w)$.

Let $a =$ the pitch ratio, and $p = aD =$ the absolute pitch. Then by the definition of the slip ratio

$$v = pn\,(1 - S) = a\,D\,n\,(1 - S) \qquad (28)$$

Substituting these values of v in equations (27), and noting that $(1 - S)^2$ times an undetermined function of S and R is merely some other undetermined function of S and R, we get the thrust and torque equations in the alternative forms:

$$\left.\begin{aligned} T &= D^2 p^2 n^2 \rho \,.\, \varphi (S, R) = D^4 n^2 \rho \,.\, a^2 \varphi (S, R) \quad \text{(a)} \\ Q &= D^3 p^2 n^2 \rho \,.\, \psi (S, R) = D^5 n^2 \rho \,.\, a^2 \psi (S, R) \quad \text{(b)} \end{aligned}\right\} (29)$$

These may be simplified in appearance by introducing a "thrust coefficient" and a "torque coefficient," but there is no general agreement in defining these coefficients, at least three different forms of definition being current.

One procedure, commonly followed by German writers, is to write the second forms of equations (29)

$$\left.\begin{aligned} T &= K_T\ D^4\ n^2\ \rho \qquad \text{(a)} \\ Q &= K_Q\ D^5\ n^2\ \rho \qquad \text{(b)} \end{aligned}\right\} \quad (30)$$

thus defining the thrust coefficient K_T and the torque coefficient K_Q by the equations

$$\left.\begin{aligned} K_T &= \frac{T}{D^4\ n^2\ \rho} = a^2\ \varphi\ (S, R) \qquad \text{(a)} \\ K_Q &= \frac{Q}{D^5\ n^2\ \rho} = a^2\ \psi\ (S, R) \qquad \text{(b)} \end{aligned}\right\} \quad (31)$$

The fact that D does not appear explicitly in the second form of equation (31, a) shows that the value of K_T, at any given values of S and R, depends only on the shape of the propeller and not on its size. Hence if K_T has been determined from a model test by substituting the observed thrust in the equation $K_T = T_1/D_1^4\ n_1^2\ \rho_1$, the same value of K_T substituted in the equation $T_0 = K_T\ D_0^4\ n_0^2\ \rho_0$ will give the thrust of the large propeller at the same slip ratio and Reynolds number, i. e. in a test that is dynamically similar to the model test.

If the effects of viscosity are negligible so that the value of R is unimportant and the Reynolds condition may be disregarded, equation (31, a) reduces to

$$K_T = f\ (S) \qquad (32)$$

a^2 being merely a dimensionless shape constant. In this case the value of K_T will be the same for all determinations at any one slip ratio, regardless of the speed of advance at which the test was made; and all values of K_T, when plotted against the values of S

at which they were determined, will give points lying along a single thrust-coefficient curve, the form of which depends only on the shape of the propeller.

When the tests at a given slip ratio are made at different speeds of advance, the values of $R = Dv/\nu$ are different; and if the effects of viscosity are not entirely negligible, tests at the same slip ratio but different speeds will not give identical values of K_T. Thus the closeness with which points determined indiscriminately at various speeds of advance lie along a single thrust-coefficient curve is a criterion for the negligibility of the Reynolds condition. If the scattering is more than can be accounted for by accidental errors of measurement, it is thereby proved that the effects of viscosity were appreciable. In that event, tests of large and small propellers can not be made satisfactorily similar merely by running them at the same slip ratio; and the impossibility of satisfying the Reynolds condition when the propellers are of very different sizes leads to the consideration of scale effects, which will be discussed a little later.

Entirely similar remarks are applicable to the torque coefficient K_Q.

The usual British practice differs from the German by preferring the absolute pitch to the pitch ratio and writing the thrust and torque equations in the forms

$$\left.\begin{aligned} T &= K'_T D^2 p^2 n^2 \rho \quad \text{(a)} \\ Q &= K'_Q D^2 p^3 n^2 \rho \quad \text{(b)} \end{aligned}\right\} (33)$$

so that the definitions of the coefficients are

$$\left.\begin{aligned} K'_T &= \frac{T}{D^2 p^2 n^2 \rho} = \varphi(S, R) \quad \text{(a)} \\ K'_Q &= \frac{Q}{D^2 p^3 n^2 \rho} = \frac{1}{a}\psi(S, R) \quad \text{(b)} \end{aligned}\right\} (34)$$

A modification of this second pair of definitions is adopted by D. W. Taylor (11) who writes, in place of equations (33),

$$\left.\begin{aligned} T &= C_T\ D^2\ p^2\ n^2 \qquad \text{(a)} \\ Q &= C_Q\ D^2\ p^3\ n^2 \qquad \text{(b)} \end{aligned}\right\} \quad (35)$$

so that

$$\left.\begin{aligned} C_T &= \frac{T}{D^2\ p^2\ n^2} = \rho\ \varphi\ (S,\ R) \qquad \text{(a)} \\ C_Q &= \frac{Q}{D^2\ p^3\ n^2} = \frac{\rho}{a}\ \psi\ (S,\ R) \qquad \text{(b)} \end{aligned}\right\} \quad (36)$$

The remarks made above in connection with K_T and the importance or unimportance of the Reynolds condition are equally applicable to

$$K'_T,\ K'_Q,\ C_T,\ \text{and}\ C_Q.$$

It will be noticed that in the first two forms of the definitions the coefficients are dimensionless, whereas in the third they have the dimensions of density. Dimensionless coefficients are virtually indispensable in correlating the results of experiments carried out in air, at various densities which can be observed but not controlled; and they have the advantage of facilitating the comparison of values expressed in English and metric units, respectively. But in the case of marine propellers we are concerned with only two, virtually constant, densities, viz. that of the fresh water in the model basin and that of sea water, which is on the average about 1.024 times as dense, the variations with temperature being insignificant in both cases. The coefficients C_T and C_Q are therefore satisfactory for use with any one set of units; and since each of them is proportional to ρ, values determined in fresh water will be correct for sea water if multiplied by 1.024.

Although the forms of the functions $\varphi\ (S,\ R)$ and $\psi\ (S,\ R)$ remain unknown until determined by experiments on propellers of the particular shape in question, they are the same throughout the equations, from (29) to (36). We therefore find, by comparing the three thrust coefficients, that

$$\varphi\ (S,\ R) = \frac{K_T}{a^2} = K'_T = \frac{C_T}{\rho} \quad (37)$$

and similarly

$$\psi\,(S, R) = \frac{K_Q}{a^2} = a\,K'_Q = \frac{a}{\rho}\,C_Q \qquad (38)$$

By means of these relations, values of the coefficients computed from the results of experiment according to any one of the definitions may be transformed so as to agree with either of the other definitions, and graphical representations of experimental values will differ only in scale.

(d) *Lift and Drag Coefficients of a Blade Element.* Let us imagine two cylindrical surfaces, coaxial with the propeller shaft and of the infinitesimally differing radii r and $r + dr$, which are less than the radius of the propeller but greater than that of the hub. Where the cylinders intersect one of the blades, they include between them a "blade element."

If this infinitely thin slice were flattened, it would be equivalent to a slice cut from a wing or airfoil by two infinitely close planes perpendicular to the span, the profile of the foil being the plane development of the profile of the blade element, and its chord being equal to the width of the blade measured on the cylinder of radius r.

A propeller blade may be regarded as consisting of a series of such elements, of continuously varying profile and width. When the propeller is running in water, the helical motion of each element with respect to the water corresponds to the rectilinear motion of the equivalent wing element with respect to the air in normal flight; and the speed and angle of attack, at which the blade element meets the water, vary continuously from root to tip.

Although the flow over a propeller blade does not take place exactly along cylindrical surfaces, the properties of each separate blade element may be determined, approximately, by testing an airfoil of the given developed profile and of finite span in the straight current of a wind tunnel. The behavior of the foil may then be described by the values of its lift and drag coefficients, and the thrust and torque coefficients of the complete propeller will evidently be related in some way to the lift and drag coefficients of its constituent elements.

The lift of an airfoil of given shape in horizontal flight depends on its size, speed, and angle of attack, and on the density and viscosity of the air. At speeds approaching that of sound, the compressibility of the air becomes important; but at ordinary flying speeds or at the still lower maximum speeds attainable in most wind tunnels, the differences of pressure set up about a wing are too small to cause any significant changes of density, and the air behaves sensibly as if it were a liquid of low but constant density. Let

l = the chord length, or width, of the foil;

A = its plan area,—if the plan is a rectangle, $A = l \times$ the span;

v = the horizontal speed relative to the air;

α = the angle of attack, or the angle between the chord and the direction of flight; and

Y = the lift, or the upward force exerted by the air on the foil.

Then we have the formal statement that

$$Y = f\ (l, v, \rho, \nu, \alpha) \qquad (39)$$

g being absent from the list of independent variables because there is no question of disturbing a free surface. And by the usual procedure this equation may be reduced to the form

$$Y = l^2 \rho v^2 f_1\ (R, \alpha) \qquad (40)$$

where $R = lv/\nu$ = the Reynolds number.

Since the shape of the foil is given, A is proportional to l^2 and we may substitute $l^2 = An/2$, where $n/2$ is a numerical shape constant. So combining n with the undetermined function and setting $nf_1\ (R, \alpha) = \varphi\ (R, \alpha)$, we may put equation (40) into the equivalent form

$$Y = A\ \frac{\rho v^2}{2} \varphi\ (R, \alpha) \qquad (41)$$

in which the form of the operator φ is fixed by the shape of the foil, as was that of f_1.

If we now introduce the abbreviations

$$\frac{\rho v^2}{2} = q \tag{42}$$

and

$$\varphi (R, a) = C_y \tag{43}$$

the lift equation takes the form

$$Y = A q C_y \tag{44}$$

The quantity q represents the impact pressure of a liquid of density ρ moving at the speed v. It is the excess of pressure over that of the surrounding atmosphere, on the leading edge of the foil at the stagnation point where the flow divides.

The dimensionless quantity C_y is the "lift coefficient" of the foil. By equations (41) and (43)

$$C_y = \frac{2 Y}{A \rho v^2} \tag{45}$$

and to determine a value of C_y, simultaneously observed values of ρ, v and Y, together with the known value of A, are to be substituted in equation (45). If this is done for a variety of speeds at each of a number of fixed angles of attack, the results may be represented by a family of curves

$$C_y = \varphi (R) \quad (a = a_1, a_2, \ldots \text{etc.}) \tag{46}$$

Or Y may be measured for various angles of attack at each of a series of fixed speeds; and if the density and viscosity of the air are constant during the measurements at each speed so that R is constant, the results may be represented by a family of curves

$$C_y = \varphi (a) \quad (R = R_1, R_2, \ldots \text{etc.}) \tag{47}$$

The numerical values of C_y obtained from equation (45) will depend on what units are used in measuring A, ρ, v, and Y. But

if the relations of these derived units to the three fundamental units of mass, length, and time are fixed by definition, changing the absolute sizes of the fundamental units does not alter the numerical values of the dimensionless quantities C_y and R. For example, the values will be the same in terms of a system based on the pound, foot and second as in terms of a system based on the meter, kilogram and second, if the two systems are "consistent" as regards the derivation of the secondary units of speed, area, etc., from the fundamental units. Similar remarks might be made with respect to the dimensionless forms of thrust and torque coefficient of a propeller.

The introduction of the factor 1/2 in passing from equation (40) to equation (41) is purely a matter of definition, its sole advantage being that it suggests a simple physical interpretation of q. British writers omit this factor, so that the 2 is absent from equation (45) and their values of C_y are half what they would be with the definition adopted here, which conforms to the usual American and German practice.

If the foil in question, or any other foil of the same shape but different size, is tested in a uniform stream of water, the value found for C_y will be the same as was found in the wind tunnel at the same angle of attack and the same value of the Reynolds number, provided that: (a) the foil is so deeply submerged that it does not disturb the free surface of the water; (b) there is no cavitation; and (c) the air and water streams are equally turbulent. This last proviso is important; for tests of an airfoil in air streams of different degrees of turbulence sometimes give considerably different results, the differences being much greater for some profiles than for others. Measurements with the foil advancing into quiet water may be expected to give somewhat different results from tests in either the wind tunnel or the water tunnel, in both of which the stream is invariably filled with fine grained turbulence. Fortunately, the thin ogival profiles commonly used for marine propeller blades appear to be less sensitive to variations of turbulence than the thicker unsymmetrical profiles used for airplane wings and airscrews.

In general, both the width and the profile of a foil may vary along its length, but to obtain values of C_y that are characteristic

of a particular profile, the foil is made uniform from end to end so that its shape is completely determined by the profile and the aspect ratio, or ratio of span to chord. If the span were infinite, the flow over the foil would be in planes perpendicular to the span and C_y would be determined by the profile alone; but with a foil of finite aspect ratio, the excess of pressure on the lower side over that on the upper causes a flow round the ends from below upward, so that a helical vortex trails away from each tip and the flow over the foil is inclined somewhat outward below and inward above, the inclination increasing from the center outward. Thus the flow is not two-dimensional except in the central plane of symmetry.

It is possible however, by approximate methods which are set forth in works on aerodynamics (12), to make an allowance for the effect of the ends and to compute, from the values of C_y obtained by testing a foil of finite aspect ratio, the values C_y would have for a foil of the same profile but infinite aspect ratio, i. e. without any free ends. It is these corrected values, which depend on the profile alone, that are of interest in connection with propeller blades.

The head resistance or drag X has the same dimensions as the lift Y; and wherever Y appears in the equations, from (39) onward, it may be replaced by X, with no other change except that the undetermined function $\varphi\ (R, \alpha)$ will have some other form, $\psi\ (R, \alpha)$. We thus obtain the equation

$$X = A\ q\ C_x \tag{48}$$

where C_x is the "drag coefficient" of the foil, defined by the equation

$$C_x = \psi\ (R, \alpha) = \frac{2X}{A\ \rho\ v^2} \tag{49}$$

All the comments made above with regard to C_y are equally applicable to C_x and need not be repeated.

For a given propeller, running in open water at a given slip and speed of advance, it is possible to compute, approximately, the speed and angle of attack at which each blade element meets the

water. If it could be assumed that each element acted independently of the others and that the flow through the propeller disc was strictly along cylindrical surfaces, the lift and drag of any element could be computed from its area, speed, angle of attack, and lift and drag coefficients. By resolving these forces along the direction of advance of the propeller and the direction of the circumferential motion of the element and adding the respective components, the contribution of this element to the thrust and torque of the propeller could be computed; and a summation over the blades would give the thrust and torque of the whole propeller.

Computations based on so highly idealized a picture of propeller action could not be expected to give quantitatively satisfactory results; but less violently simplified methods, based on the consideration of the properties of separate blade elements, have nevertheless been reasonably successful in predicting the behavior of propellers. Any discussion of blade-element theories would be far beyond the scope of this paper, but without going into such details it will be evident that there is an intimate connection between the properties of the separate blade elements and those of the complete blade, and that the study of the behavior of airfoils or aquafoils is indispensable to any deeper understanding of the much more complicated phenomena of propeller action.

(e) *Scale Effect in the Case of Airfoils.* Returning to equation (41), confining our attention to a particular angle of attack, and comparing the lift of an airplane wing of chord l_0 with that of a model foil of chord l_1, we have the relation

$$\frac{Y_0}{Y_1} = \frac{A_0 \rho_0 v_0^2 \varphi(R_0)}{A_1 \rho_1 v_1^2 \varphi(R_1)} \qquad (\alpha \text{ const.}) \qquad (50)$$

where

$$R = \frac{l v}{\nu} = \frac{l v \rho}{\mu} \qquad (51)$$

μ denoting the ordinary, absolute viscosity, given in the physical tables.

To make the model test dynamically similar to the full scale flight and convert equation (50) into a definite statement, it is necessary to make $R_1 = R_0$, or

$$\frac{l_1 v_1 \rho_1}{\mu_1} = \frac{l_0 v_0 \rho_0}{\mu_0} \tag{52}$$

The viscosity of air is sensibly independent of pressure and varies only rather slowly with temperature, so that μ_1 will not differ much from μ_0. Any difference is likely to be in the direction $\mu_1 > \mu_0$. As a rule, the highest speeds available in wind tunnels are somewhat less than the speeds of airplanes in flight; and except in some of the very large modern tunnels, l_1 has to be a small fraction of l_0. Consequently, the only way in which equation (52) can be satisfied is by making the density ρ_1 in the wind tunnel much greater than the density ρ_0 of the atmosphere in which the full sized plane is to fly. This is possible in the modern high-pressure tunnels, and when the pressure is so adjusted as to satisfy equation (52), equation (50) takes the definite form

$$\frac{Y_0}{Y_1} = \frac{A_0 \rho_0 v_0^2}{A_1 \rho_1 v_1^2} \quad (a \text{ const.}) \tag{53}$$

But until the advent of the high-pressure tunnel this resort to an increase of density was not available and it was impossible to satisfy the Reynolds condition for similarity.

If the form of the curve

$$C_y = \varphi(R) \tag{54}$$

for the given value of α were known over a sufficient range of R, the value of $[\varphi(R_0)/\varphi(R_1)]$ in equation (50) could be found from it, and Y_0 could be computed from the observed Y_1 without its being necessary to satisfy the Reynolds condition. A short piece of the curve can be determined by testing the model at various speeds in a wind tunnel working at atmospheric pressure, but in the absence of any valid theoretical reason for assigning a particular form to equation (54), extrapolation to the much higher values of R that occur in full scale flight is entirely unreliable. Similar re-

marks might be made with respect to the drag coefficient, and these variations of C_y and C_x over large ranges of R, which were unpredictable in the early days of wind tunnel research, received the name "scale effect."

The same term is often used to designate the variations, with Reynolds number, of the characteristic dimensionless coefficients which occur in the study of other phenomena of fluid motion, such as the resistance coefficient of smooth spheres moving through a fluid or the coefficient of skin friction of a liquid flowing along a straight pipe.

Such variations are usually slow and regular over considerable ranges of the appropriate Reynolds number—often over the whole range of R that has hitherto been accessible to experiment. In such a case, if C is the coefficient in question and R is the appropriate Reynolds number, the curve

$$C = \varphi\ (R) \tag{55}$$

determined by experiment will be of some fairly simple form, with no discontinuities or sudden changes of slope. Occasionally however a limited "critical" range is found, within which the value of C varies much more rapidly with R than on either side, so that the complete curve consists of two pieces, of simple though perhaps different shapes, connected by a short transition piece which is of a very different shape from either of the others. Such a critical change usually—probably always—corresponds to a radical change in the nature or configuration of the fluid motion.

The most familiar instance of this is the rapid increase of the coefficient of pipe friction as R increases through the range 2000 to 3000 ($R = Dv/\nu$; D = diameter of the pipe; v = mean speed of flow along it; all quantities measured in terms of any system of units that is consistent with the C. G. S. system). This change is due to a change from stream-line to turbulent flow and proceeds as turbulence develops. It may, with sufficient care, be made to take place gradually, and the form of the transition curve is fairly well known; but if the critical range is approached from lower values of R by increasing the speed of flow, the onset of turbulence is usually so sudden that the increase of the resistance appears to be discontinuous.

Another example is afforded by the resistance coefficient C of a smooth sphere of diameter D moving through a fluid of density ρ and kinematic viscosity ν, at a speed v that is not over half the speed of sound in the fluid. The resistance X may be expressed by the equation

$$X = AqC$$

where $A = \pi D^2/4$, $q = \rho v^2/2$, and $C = \varphi\ (Dv/\nu) = \varphi\ (R)$.

If the test is in quiet water or in a wind stream that is as nearly as practicable free from turbulence, there is a critical range in the vicinity of $R = 250000$, within which as R increases C decreases to about one-third of its previous value, whereas on both sides of the critical range the variation of C with R is very much slower. Increasing the turbulence of the fluid causes the change to start earlier and take place more gradually, the critical range being wider and the value of C after the rapid change has ended being somewhat lower than with less turbulence.

For some of the blade-element profiles that occur in marine propellers, the curves of lift and drag coefficients against R have been found to have similar ranges of unusually rapid change, and these are of interest in connection with scale effects on the thrust and torque coefficients of propellers.

(f) *Scale Effect on Marine Propellers.* We return to the thrust and torque equations in the form (30), or

$$\left.\begin{aligned} T &= D^4 n^2 \rho K_T \\ Q &= D^5 n^2 \rho K_Q \end{aligned}\right\} \tag{56}$$

where

$$\left.\begin{aligned} K_T &= \varphi\ (S, R) \\ K_Q &= \psi\ (S, R) \end{aligned}\right\} \tag{57}$$

the shape factor a^2 having been combined with the shape functions φ and ψ, as a matter of convenience.

With propellers running in water, it is impossible to satisfy the Reynolds condition for similarity between tests of geometrically

similar propellers of very different diameters, D_0 and D_1. The slip condition may be fulfilled by adjusting v_1 or n_1 so that

$$\frac{v_1}{v_0} = \frac{D_1 n_1}{D_0 n_0} \tag{58}$$

but after this has been attended to there remains the question of the shapes of the curves

$$\left.\begin{aligned} K_T &= \varphi(R) \quad (S \text{ const.}) \qquad \text{(a)} \\ K_Q &= \psi(R) \quad \text{“} \qquad\qquad \text{(b)} \end{aligned}\right\} \tag{59}$$

for the given value of the slip ratio,

$$S = 1 - \frac{v_0}{a D_0 n_0} \tag{60}$$

Let the model be run in open water in the basin, and let its thrust and torque be measured over the whole available range of carriage speeds, while n is kept proportional to v so that S remains constant. Let the observed values of T, corrected for the resistance of the bare hub, be plotted over those of n^2 or v^2. In equations (56) D and ρ are constants, and if K_T is also constant, the points will lie along a straight line through the origin. Conversely, if the points do lie in this way, it is thereby proved that K_T was constant and that there was no scale effect on the thrust within the range of R covered by the variations of v, with D and ν constant. If, on the other hand, the points do not fall on a straight line within the precision of the measurements, K_T must have varied with R; and by plotting values of K_T computed from the equation $K_T = T/D^4 n^2 \rho$, the shape of the curve (59, a) can be determined over the given range of R. Similarly with K_Q.

The experimental range of R may be somewhat extended by using larger or smaller models, but with any ordinary model basin equipment it is impossible to come anywhere near the values of R for large propellers under their normal working conditions, and only the lower part of the curve can be determined.

Experience with other cases of fluid motion, and elementary physical considerations, lead one to expect that scale effects of all sorts, which are always manifestations of the action of viscous forces, will tend to decrease at very high Reynolds numbers; but only experiment can show how high. In the present case, the success with which the performance of large propellers has been predicted from the results of tests of 16-inch models indicates that the values of R in the model tests were high enough to be above the range where scale effects might be of serious importance. The accuracy with which predictions from model tests can be checked is limited by the errors in measuring thrust and torque in ship trials, and when higher accuracy has been attained in these measurements some appreciable scale effects may come to light. All that can be said at present is that the scale effects appear to be small in comparison with other uncertainties in interpreting the results of open tests of model propellers,—especially the uncertainty in the wake fraction.

But between 14-inch and 5-inch propellers, running in open water at corresponding speeds, marked scale effects have been observed; and predictions of full-scale performance from tests of too small models are likely to be considerably in error. The systematic investigation of these scale effects has only recently been undertaken by experimenters; and since the subject is still in an early stage of development, only a few comments on some of the results will be given here. For details, the reader may be referred to the papers by Gutsche (13) and Allan (14) in which their experiments are described.

The investigators just mentioned start by studying scale effects on the lift and drag coefficients of blade-element profiles, and locating critical regions within which the coefficients change very rapidly or even abruptly, with small changes of the Reynolds number. In this connection, the Reynolds number may be defined as $R_b = l\ v_b/\nu$, where l = the chord of the element and v_b = its speed relative to the water. In the complete propeller, when advancing at the speed v and constant slip, the relative speed v_b of an element at radius r is proportional to vr; and the angle of attack is fixed by the slip ratio.

The most interesting of the general conclusions to be drawn from the investigations is somewhat as follows: When a small model propeller is running at low speed in quiet water, all elements of its blades may be working at values of their Reynolds numbers below their critical ranges, the flow over the blades being, presumably, in stream lines and free from turbulence. If the speed is increased at constant slip, a critical change of regime may start at the tips where the speed is highest; and as the speed is raised farther, the critical change may work inward toward the hub until the whole blade is running in the supercritical region of R_b. The change of regime at any given radius and the changes in the lift and drag coefficients of the blade element there may be rapid or even sudden; but they occur on only a very small fraction of the whole area of the blade, and the effect on the thrust and torque coefficients will be gradual. Nevertheless, the scale effect on K_T and K_Q may be much greater than if no such critical changes were going on anywhere on the blades.

With a larger model at correspondingly higher speeds, the whole of each blade may be working in the supercritical region; or if any critical change is still possible, it may be on the less important sections near the hub, where even large changes in C_x and C_y could not have any great effect on the values of K_T and K_Q. If the whole propeller is working in its supercritical state, any further increase of R, whether by increasing the speed or by passing to a larger model, will not lead to any further critical phenomena; and experience seems to show that in the supercritical state scale effects on propeller performance are small. It just happens that with propellers of the usual shapes, critical changes and large scale effects are likely to occur in tests of models as small as 5 or 6 inches in diameter, but not with diameters as large as 14 inches.

The investigations cited are not yet adequate to the accurate prediction of scale effects over the large range of R up to full scale operation; and the influence of turbulence, which may be expected to diminish scale effects over any given range of R, remains to be determined. But they throw light on the results of earlier experience, which show that it is desirable to make propeller tests on as large models as is economically practicable.

III. CAVITATION.

7. *Preliminary Remarks.* If a solid body is held fixed in a stream of liquid, its presence influences the velocity field in its vicinity. At places where the lines of flow are crowded together by the obstruction, the speed is increased and the static pressure falls,—in general agreement with Bernoulli's theorem. And wherever the liquid is forced to flow along a curved path, as over the convex back of a propeller blade or where an eddy is set up behind an obstruction, centrifugal force tends to throw the liquid away from the center of curvature and so lowers the static pressure on the concave side of the stream. Either of these causes or both acting together may lower the static pressure at some point in the liquid so far that the liquid can boil there; and if this happens, an open space filled with vapor, or a "cavity," will be formed in the body of the liquid.

As usual, only the relative motion is of importance, and the effects will be the same whether the body is fixed in a stream or moves through an otherwise undisturbed body of liquid.

In some circumstances, a stable cavity of considerable size may be formed and persist in a fixed position with respect to a submerged body. This is illustrated by the helical eddies sometimes seen trailing from the tips of propeller blades. Tip vortices are always present when the pressure on the face of the blade near the tip is greater than on the back, although, if the water is clear, they are not visible when the thrust is low. But if the flow round the tip from face to back is rapid enough, the centrifugal force in the resulting vortex lowers the pressure at the axis so far that the water can boil. The core of the vortex is then filled with vapor and a hollow, tubular eddy appears, in the form of a helix attached to the blade tip. As the eddy trails away from its point of origin close to the blade, its kinetic energy is gradually dissipated by viscosity until the speed of rotation is no longer sufficient to keep the pressure in the core down to the vapor pressure of the water. The core then begins to break up, first into "sausages" and then into a string of spherical bubbles, which finally collapse and disappear still farther aft.

A more important type of cavitation consists in the continual formation and disappearance of small, isolated cavities, i.e., bubbles of vapor, within a more or less definitely bounded region or "cavitation field,"* the bubbles forming at the upstream side of the field, passing through it, and collapsing at the downstream side. This is exemplified in the "burbling" cavitation which occurs on the convex backs of propeller blades.

If the speed and thrust of a model propeller are gradually increased, cavitation is likely to begin on the back of each blade with the appearance of small bubbles near the line of maximum blade thickness. As this incipient boiling becomes more active, it occurs over a wider strip of the surface. The individual bubbles last longer and are swept along before collapsing, so that a cavitation field is formed on the back of the blade. And as the speed of the water relative to the blade continues to increase, the cavitation field spreads,—forward to some extent, but mainly toward the trailing edge or even beyond it. The bubbles do not, however, all move as fast as the water outside the field. Some at least creep along the surface so slowly that, by stroboscopic methods, a particular bubble may be seen on the blade during several revolutions of the propeller. It appears, from the observations of H. Mueller (15) on foils of uniform section held in a stream of water, that these bubbles are roughly hemispherical, with their flat bases on the surface of the foil or blade.

If the bubbles collapse before passing off the blade, each collapse causes a violent water hammer on a minute area of the surface, and this sort of hammering action, wherever it may occur on a solid surface, appears to be the principal cause of erosion (16, 17, 18).

In discussing propeller cavitation it may be assumed that the static pressure in the water immediately surrounding a cavity is equal to the pressure of saturated water vapor at the given temperature. The pressure inside may be slightly increased by the liberation of dissolved air or other gases; and the static pressure of the water outside is assisted in balancing the pressure inside the cavity by a small compressive effect of surface tension. Moreover, if the

* This useful descriptive term is due to Commander Lybrand P. Smith, U. S. N. See reference 10 in the bibliography.

cavity or bubble is expanding or contracting, the balance is not exact. But these are all minor matters, and in practice the assumption mentioned is sufficiently accurate.

So long as a propeller is not cavitating, its performance does not depend on the pressure to which it is subject; for the only effect of changing the pressure is to change the density and viscosity of the water by negligible amounts. But any change of the atmospheric pressure on the free surface produces an equal change of pressure at every point in the water,—inside the propeller race as well as outside. Hence if a propeller is running at a speed and thrust that are nearly, but not quite, high enough to bring the static pressure at some point on the blades down to the vapor pressure and start cavitation there, lowering the air pressure will make the propeller cavitate; or if it is already cavitating, the cavities can be squeezed out of existence by raising the air pressure on the water surface.

When a model propeller is run in the basin at the designed submergence, the head of water over it is reduced from that over the full sized propeller, in the same ratio as the diameter; but the pressure of the atmosphere, equivalent to a head of about 33 feet of sea water, is not reduced. The model is under a relatively much higher absolute pressure than the large propeller, and if the conditions that suffice for dynamical similarity in the absence of cavitation are fulfilled, cavitation on the model is delayed to a much higher speed and thrust than correspond to the development of cavitation on the large propeller. Open-water tests of a model are therefore not capable of showing when the large propeller will begin to cavitate or how its thrust and torque will be affected as the cavitation increases. For this purpose it is necessary to test the model under reduced pressure, and this has led to the construction of closed-circuit water tunnels, in which the propeller rotates in a fixed position while the water flows past it, and the air pressure on a free water surface at the highest point of the circuit can be reduced by means of an air pump.

The earliest successful tunnel of this sort seems to have been the one built by Sir Charles Parsons in 1910, which was first publicly described, in 1932, by Mr. S. S. Cook in his discussion of a paper on cavitation by Messrs. G. Kempf and H. Lerbs (19). The

more recent tunnels at the Washington Navy Yard and at the Hamburg Schiffbau-Versuchsanstalt have been fully described by Commander H. E. Saunders, (C.C.), U.S.N. (20) and by Mr. H. Lerbs (21), respectively. In each case, the propeller shaft passes through a gland in the curved wall of the tunnel, the thrust and torque dynamometers being outside; and glass windows are provided, through which the propeller can be observed in action by stroboscopic methods. A presumably somewhat similar tunnel exists in Paris but the present writer has not seen any description of it.

With the air pressure constant, cavitation may be induced or eliminated by varying the speed of the water stream or the rate of rotation of the propeller or both. Or with the speed and slip held constant, lowering and raising the air pressure affords an alternative possibility for studying how, where, and when cavitation occurs on propellers of various shapes. Virtually all the information hitherto published regarding the details of propeller cavitation has been obtained from experiments with closed-circuit, variable-pressure water tunnels.

8. *Notation and Definitions.* For the discussion of cavitation, the notation hitherto employed may be supplemented as follows:

h = the depth of the center of the propeller below the free surface of the water.

p = the absolute pressure at the depth h, outside the propeller race in an open water test, or in the uniform stream of the water tunnel as it approaches the propeller; $p = g\,\rho\,h +$ the presure of the air on the free surface.

e = the vapor pressure of the water at the working temperature.

$P = p - e$ = the amount by which the absolute pressure at the depth h exceeds the vapor pressure.

Δ = the amount by which the action of the propeller on the water reduces the static pressure at any point, below the pressure at that level in the water outside the influence of the propeller.

$q = \rho v^2/2$ = the impact pressure due to the speed of advance; in the water tunnel, v is the speed of the uniform stream approaching the propeller.

$I = P/q$ = the cavitation index.

9. *The Independent Variables.* Under ideally steady conditions a propeller blade might cavitate more when at the top of its path than when lower down and under a higher pressure; but no such difference is observable in the water tunnel, and we may as well ignore the vertical variations of hydrostatic pressure across the propeller circle.

On this basis, the absolute static pressure at any point in the water may be taken to be $(p - \Delta)$, and so long as this is everywhere greater than the vapor pressure e, there can be no cavitation. But with increasing speed and thrust, the speed of the water relative to the blades increases; and wherever the pressure has already been lowered, it falls still farther. If this goes on until $\Delta = p - e$ at some point, i.e. until $p - \Delta = e$, cavitation will begin there.

Heating the water increases its vapor pressure and so diminishes the depression Δ needed to initiate cavitation. Hence, other things being equal, a propeller will cavitate earlier in warm water than in cold. But there is no obvious reason, either experimental or theoretical, for supposing that the value of e is of any importance except as it affects the value of $(p - e)$ and serves as a zero point from which the value of $P = p - e$ is to be measured. In the absence of any such reason it is legitimate to assume that the value of P is an adequate specification of the conditions of pressure which affect cavitation. And if we disregard the effects of surface tension as insignificant, P is the only new independent variable to be added to the earlier list, which sufficed when cavitation was expressly excluded from the discussion.

The model propellers tested in the Washington and Hamburg tunnels are usually between 6 ins. and 10 ins. in diameter, and from what was said in section 6,f it would seem that propellers as small as these might show considerable scale effects if tested in quiet water. It is to be expected, however, that in the turbulent stream of the water tunnel the influence of viscosity and the resulting scale effects will be less than in quiet water; and experience at Hamburg is said (22) to have shown that they were of no serious importance over the range of Reynolds numbers covered in the tests. The absence of any mention of difficulties with scale effects by the authorities of the Washington tunnel points in the same

direction; and the presumption is that scale effects will continue to be small up to full scale values of R, in comparison with the errors in measuring thrust and torque during ship trials. Future improvements in the accuracy of such measurements may show that scale effects are not entirely negligible, but for the present they may be ignored, and we shall pay no further attention to the Reynolds condition.

This is equivalent to omitting the viscosity from the list of independent variables; so with the omission of ν from the list (II) of section 1 and the addition of the new variable P, the list is

$$D, v, S, \rho, g, P \qquad (61)$$

in which P has the dimensions of a pressure.

10. *The Conditions for Dynamical Similarity when Scale Effects are Negligible but Cavitation may Occur.* There are still 6 independent variables, and 3 independent dimensionless Π's are possible. And since $q = \rho v^2/2$ is a pressure, the ratio $P/q = I$ may be taken as the third combination, to go with S and F. Hence if two propellers, of the same shape and of diameters D_0 and $D_1 = rD_0$, are tested under geometrically similar conditions of submergence and wake, and if scale effects are negligible, the tests will be dynamically similar if they are so arranged that the equations

$$\left.\begin{aligned} S_1 &= S_0 \qquad (a) \\ F_1 &= F_0 \qquad (b) \\ I_1 &= I_0 \qquad (c) \end{aligned}\right\} (62)$$

are satisfied.

The meaning and justification of this statement follow from considerations such as were set forth in section 3 and need not be discussed in detail.

The new requirement expressed by equation (62,c) may be called the "pressure condition."

For practical use, the conditions must be expanded into more convenient forms. The slip ratio cannot be measured directly but has to be computed by means of the equation

$$S = 1 - \frac{v}{a\,D\,n} \tag{63}$$

And since the pitch ratio a is the same for both propellers, equation (62,a) is equivalent to

$$\frac{v_1}{D_1\,n_1} = \frac{v_0}{D_0\,n_0} \quad \text{or} \quad \frac{n_1}{n_0} = \frac{v_1\,D_0}{v_0\,D_1} \tag{64}$$

Equations (62,b) and (62,c) are abbreviations for

$$\frac{D_1\,g_1}{{v_1}^2} = \frac{D_0\,g_0}{{v_0}^2} \quad \text{or} \quad \frac{v_1}{v_0} = \sqrt{\frac{D_1\,g_1}{D_0\,g_0}} \tag{65}$$

and

$$\frac{2\,P_1}{\rho_1\,{v_1}^2} = \frac{2\,P_0}{\rho_0\,{v_0}^2} \quad \text{or} \quad \frac{P_1}{P_0} = \frac{\rho_1\,{v_1}^2}{\rho_0\,{v_0}^2} \tag{66}$$

As usual, we set $g_1 = g_0$ and $D_1/D_0 = r$; and to simplify matters we shall suppose that $\rho_1 = \rho_0$. Equations (64, 65, 66) then assume the forms

$$\left.\begin{aligned} \frac{n_1}{n_0} &= \frac{1}{r} \cdot \frac{v_1}{v_0} && \text{(a)} \\ \frac{v_1}{v_0} &= \sqrt{r} && \text{(b)} \\ \frac{P_1}{P_0} &= \frac{{v_1}^2}{{v_0}^2} && \text{(c)} \end{aligned}\right\} \tag{67}$$

which are equivalent to equations (62) if both propellers are run in water of the same density. When the model test in fresh water is so arranged that n_1, v_1, and P_1 satisfy equations (67), full-scale values of thrust and torque predicted from the results of the model test will be correct for the large propeller running in fresh water, if scale effects are negligible and the geometrical conditions of submergence and wake, or absence of wake, are similar. The values for sea water will then be 1.024 times the fresh-water values.

The condition of equal slip ratios is evidently of fundamental importance and must always be satisfied. So also must the pressure condition if there is any danger of cavitation, although if both pressures are high enough to prevent cavitation, it makes no difference what they are. But although the Froude condition of corresponding speeds is of prime importance in testing a model hull, either alone or in combination with its propeller, it is of much less importance when the propeller is tested separately.

In the absence of cavitation, the weight of the water can influence the behavior of a propeller in an open test, only in consequence of alterations of the surface level by the action of the propeller; and, as noted in section 6,b, the Froude condition may nearly always be disregarded in open tests of propellers that are not cavitating, without thereby invalidating the transfer of thrust and torque coefficients from model to full-scale runs. But since the absolute static pressure at shaft level and the value of P depend on the weight of the water as well as on the air pressure, fulfillment of the Froude condition would seem to be essential to dynamical similarity after cavitation has set in.

Nevertheless, it is stated by Mr. Lerbs (22) that experiments in the Hamburg tunnel have given the unexpected result that departures from the Froude condition are unimportant (nicht wesentlich), and that it is permissible to determine thrust and torque coefficients for use in full-scale computations without having to measure the inconveniently low thrusts and torques that would be exerted by the model at the low speeds demanded by the Froude condition.

In this tunnel, the stream past the propeller station is confined by a tube and the free surface on which the reduced air pressure acts is not directly over the propeller but some distance away. This construction eliminates the possibility of surface disturbances, and so long as there is no cavitation, the weight of the water and the Froude condition do not come into the discussion at all; but it is not evident why the Froude condition should be of no importance even after cavitation has developed. However, the observed fact justifies the experimental procedure adopted at Hamburg, which consists in keeping the water speed v_1 at some convenient constant value, regardless of equation (67,b), and adjusting P_1 so as to

satisfy equation (67,c) for each value of v_0 for which the thrust and torque coefficients are desired.

Let $A =$ the head of water equivalent to the air pressure on the free surface, and let $V =$ the head equivalent to the vapor pressure e. Then at the depth h we have, in general,

$$P = g\rho\,(h + A - V) \tag{68}$$

and to satisfy equation (67,c) we must make

$$\frac{P_1}{P_0} = \frac{h_1 + A_1 - V_1}{h_0 + A_0 - V_0} = \frac{v_1^2}{v_0^2}$$

or

$$A_1 = (h_0 + A_0 - V_0)\,\frac{v_1^2}{v_0^2} - h_1 + V_1 \tag{69}$$

In this equation, the values of h_0, A_0, and V_0 are fixed by the working conditions of the large propeller; h_1 is the depth of the center of the model below the free surface in the tunnel; V_1, like V_0, may be found from the steam tables; and v_1 is the constant water speed in the tunnel. Substitution of these values puts equation (69) into the form

$$A_1 = \frac{M}{v_0^2} - N \tag{70}$$

where M and N are known constants. For any given speed of advance of the large propeller, substitution of v_0 in this equation gives the equivalent head A_1 to which the air pressure in the tunnel must be reduced in order to satisfy the pressure condition. When this adjustment has been made, the small and large propellers, running at the same slip ratio, will cavitate in the same way and to the same extent, and thrust and torque coefficients determined in the model test will be applicable to the full-sized propeller.

On the other hand, if all three conditions for dynamical similarity are satisfied, equations (67) assume the simpler and more restrictive forms

$$\left.\begin{aligned} \frac{n_1}{n_0} &= \frac{1}{\sqrt{r}} && \text{(a)} \\ \frac{v_1}{v_0} &= \sqrt{r} && \text{(b)} \\ \frac{P_1}{P_0} &= r && \text{(c)} \end{aligned}\right\} \quad (71)$$

and equation (69) takes the form

$$A_1 = r\ (h_0 + A_0 - V_0) - h_1 + V_1$$

which fixes the value of A_1 and therefore, by equation (68), that of P_1.

The standard practice at the Washington tunnel (23,24) is to use the equations in this form, working with a constant pressure in the tunnel and adjusting v_1 and n_1, in accordance with equations (71,a) and (71,b), to the values of v_0 and n_0 for which information is desired.

11. *Remarks on the Wake Fraction.* In the equations of the preceding section, v_1 denotes the speed of the stream in the water tunnel and v_0 must therefore be the true speed of advance of the large propeller behind the ship,—so far as the adjective "true" has any definite meaning in this connection. If v_s is the speed of the ship and w_0 is its wake fraction, we must make

$$v_0 = (1 - w_0)\ v_s \qquad (72)$$

and the comparison is between the performance of the model in the tunnel at the water speed v_1, and that of the large propeller when driving the ship at the speed v_s. The question is, what value to use for w_0 in adjusting v_1 to a given v_s.

Let w_1 be the wake fraction for the combination of model hull and propeller; and let us suppose, temporarily, that there is no scale effect on the wake fraction so that $w_0 = w_1$: then we have to consider how to determine w_1. For the conditions under which the model propeller works in the turbulent and confined stream of the water tunnel are not the same as when it is advancing through quiet water, and it cannot be assumed, offhand, that a wake frac-

tion determined from runs in the basin will be correct for use in adjusting the water speed in the tunnel.

The assumption may be tested by measuring the thrust at a given rate of rotation, first at the speed of advance v in open water, and second at the water speed v in the tunnel without reducing the air pressure. If the thrust is the same in both cases, the wake fraction found from runs in the basin may be used for tests in the tunnel.

If the difference is too great to be ignored, the wake fraction for the tunnel may be determined directly. Let T be the thrust measured when the propeller is running behind its hull in the basin, at the rate of rotation n and at the hull speed v . And let v_1 be the water speed at which the propeller gives the thrust T, when turning at the rate n in the tunnel. Then the wake fraction is

$$w_1 = 1 - \frac{v_1}{v_m}$$

Whether such a determination is needed will depend on the characteristics of the tunnel and on the ratio of the diameter of the propeller to the diameter of the jet or stream in which it works. From statements in a paper by Lt. Commander C. O. Kell, (C.C.), U.S.N. (24) it appears that in the Washington tunnel and for propellers of the sizes usually tested, direct determinations of w_1 are unnecessary, and that values determined from tests in the basin may be used.

It must be noted, however, that comparisons between runs in the basin and runs in the tunnel at atmospheric pressure do not cover the case of cavitation at reduced pressure. The question remains, whether a determination of a wake fraction for a given combination of hull and propeller from runs in the model basin would give the same value if the barometric pressure could be reduced so far as to make the propeller cavitate. It does not seem likely that the flow of water in the wake of a hull ahead of the propeller will be much influenced by cavitation on the propeller, or that the value of the wake fraction will be seriously affected. But some doubt on this point is unavoidable.

When this is considered, in connection with the uncertainty of the scale effect to be allowed for in passing from w_1 to w_0, it is evident

that the choice of a suitable value for w_0 requires the exercise of a good deal of judgment based on experience—a euphemism for expert guesswork. But it may be remarked that a " suitable " value is simply a value that makes predictions from model experiments turn out to be confirmed by measurements taken during ship trials. And in view of the scarcity of accurate trial run data there is still a good deal of leeway in the meaning of " suitable."

12. *Thrust and Torque Coefficients.* The method followed in section 3, when applied to the list of variables (61), gives thrust and torque equations which differ from equations (11) and (14) only by the substitution of the cavitation index I, which refers to possible cavitation, for the Reynolds number R, which referred to the scale effects now treated as negligible. And if v is replaced by n, as was done in section 6,c, we get the equations in the forms

$$\left.\begin{aligned} T &= K_T D^4 n^2 \rho \\ Q &= K_Q D^5 n^2 \rho \end{aligned}\right\} \quad (73)$$

where

$$\left.\begin{aligned} K_T &= \varphi\ (S,\ F,\ I) \\ K_Q &= \psi\ (S,\ \mathrm{F},\ I) \end{aligned}\right\} \quad (74)$$

Thrust and torque coefficients defined in either of the other two ways will be connected with K_T and K_Q by equations (37) and (38), as before.

On the supposition that the Froude condition may be disregarded, as was found to be the case in the Hamburg tunnel, or that the condition is always satisfied, as in the standard practice at the Washington tunnel, equations (74) reduce to

$$\left.\begin{aligned} K_T &= \varphi\ (S,\ I) \\ K_Q &= \psi\ (S,\ I) \end{aligned}\right\} \quad (75)$$

The definition of the cavitation index is

$$I = \frac{P}{q} = \frac{2P}{\rho v^2}$$

With the propeller running at a constant speed and rate of rotation, cavitation cannot occur if P is higher than some definite value. Or if P is held constant while v and n are increased proportionally so that S remains constant, cavitation does not begin until v has attained some definite value. For each value of the slip ratio S, the cavitation index has a certain critical value $I_c = f(S)$, such that if $I > I_c$ cavitation is impossible. If $I = I_c$, cavitation is on the point of starting; and if I is decreased farther, either by lowering the pressure or by raising the speed, cavitation increases.

Under the assumed circumstances—scale effects negligible and Froude condition either satisfied or negligible—when $I > I_c$, its value is of no further importance and equations (75) reduce to

$$\left.\begin{aligned} K_T &= \varphi\,(S) \quad (I > I_c) \\ K_Q &= \psi\,(S) \quad\quad " \end{aligned}\right\} \tag{76}$$

Equations (31) which were obtained by excluding cavitation, from the start, are equivalent to equations (76) when R is omitted. They are merely special cases of the more general equations (75), which hold for all values of I when scale effects are negligible.

13. *Comparison of a Blade Element with an Aquafoil.* Burbling, back cavitation is important, and in many cases, one object of the designer will be to delay back cavitation to as high a thrust as possible. The ability of a propeller to deliver a high thrust without cavitating seriously and absorbing an excessive amount of power depends on the blade profiles, among other things; and the selection of suitable profiles is facilitated if the designer has at his disposal information concerning the lift and drag coefficients and relative sensitiveness to cavitation, of various profiles. Such information may be obtained by testing aquafoils fixed at various angles of attack in a variable-pressure water tunnel. Varying the angle of attack corresponds to varying the slip ratio of a propeller, and the speed of the stream in the tunnel corresponds to the relative speed along its helical path, at which a blade element of the rotating propeller meets the water.

If v_b is the speed of the water past the foil in the tunnel and $q_b = \rho v_b^2/2$ is the resulting impact pressure, the cavitation index may be defined as

$$I_b = \frac{P}{q_b}$$

where P has its usual meaning.

For any given angle of attack α, when I_b has a sufficiently high value no cavitation can occur; and if the effects of surface disturbance and viscous forces are negligible, the lift and drag coefficients of the foil will depend only on α. But if I_b is less than a certain critical value, which depends on α as well as on the shape of the profile, the foil will cavitate and its lift and drag coefficients will depend on both α and I_b.

For a blade element of a complete propeller, at a given radius, the relative speed v_b and the angle of attack α depend on the speed of advance v, the rate of rotation n, and the pitch ratio at the given radius; and their values can be computed, by the aid of somewhat shaky assumptions. The value of I_b can then be computed from the known value of P; and the behavior of the blade as regards cavitation at the given radius may be expected to be more or less like the behavior of a foil in the water tunnel, if the foil has the same profile as the blade at the given radius, if it is set at the angle of attack computed from the given slip and pitch ratios, and if the cavitation index I_b has the same value for the foil as for the blade element.

It is not to be expected that very accurate conclusions about the behavior of a cavitating propeller can be drawn in this way from experiments on foils. Nevertheless, such comparisons are of some interest and may, in time, prove to be of considerable value in suggesting desirable blade profiles. But the subject can only be hinted at here, and for further information and references the reader may consult a paper by Lieutenant Commander C. O. Kell (24).

The reader who has followed the foregoing discussion of some of the questions which must be considered in planning experiments on model screw propellers and interpreting the results will have perceived that while the approximate theory which serves as a guide is quite simple, there are many practical difficulties in folfowing the guide. And he will probably have been impressed, as the writer has been, with respect for the technical experts who can find their way through such a maze of approximations and assumptions to results which are of great practical value.

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MARINE REFRIGERATION.

By Henry Edwin Bethon.*

INTRODUCTION.

The purpose of this paper is to discuss refrigeration as currently applied to cold storage, air conditioning, process water cooling and ship's service store equipment aboard Naval vessels. The refrigerant used in the case of all Naval vessels under construction is dichlorodifluoromethane, commonly known as "Freon-12" and therefore this paper will discuss only "Freon-12" refrigerant installations based on direct expansion of the refrigerant.

In recent installations, the cold storage compartments were cooled by the direct expansion of the refrigerant but the air conditioning of certain special compartments was accomplished by the indirect expansion method, wherein the recirculation of brine is employed. This method of air conditioning is now being supplanted by entirely independent air conditioning units installed throughout the vessel, as may be required.

Prior to the use of "Freon-12," the refrigerants in common use aboard Naval vessels were ethyl chloride, sulphur dioxide and carbon dioxide. Carbon dioxide is still used in the refrigerating plants aboard all of the present capital ships and in many cruisers built before the year 1933. In the carbon dioxide installations, the refrigerant cools the brine, which in turn is recirculated to and from the cold storage and air conditioned compartments.

The size of the refrigerating plant for the cold storage compartments varies with the complement and type of vessel. The following table gives the size of the refrigerating plant installations aboard various types of vessels:

* Associate Engineer, Design Division, Bureau of Engineering.

Ship	*Number and size of machines*
Aircraft Carrier	3- 6 ton units
Battleship	3- 3 ton units
Cruiser	2- 3 ton units
Destroyer	2- 1 ton units
Submarine	1- ½ ton unit

The present application of air conditioning aboard ship merely constitutes the heating or cooling of the air for the purpose of rendering the quarters more comfortable or to keep various compartments within certain temperature limits.

PART I.

PROPERTIES OF REFRIGERANTS.

Definition:—Refrigerants are heat carrying mediums which, during their cycle absorb heat at a low temperature level, are compressed by a heat pump to a higher temperature where they are able to discharge the absorbed heat together with that added during the compression to the condenser cooling water or circulating air. The ideal refrigerant would be one which could discharge to the condenser all the heat which it is capable of absorbing in the evaporator or cooler. All refrigerating mediums, however, carry a certain portion of the heat from the condenser back to the evaporator and this reduces the heat absorbing capacity of the medium on the low side of the system.

Classification:—The various types of refrigerants may be classified as follows:

Class (1) The four (4) main refrigerants of the past twenty years or more are ammonia, carbon dioxide, methyl chloride and sulphur dioxide.

Class (2) Refrigerants which are liquid at normal temperature or, in other words, high boiling point refrigerants such as methylene chloride (Carrene) and others. These refrigerants find their chief use in centrifugal compression. In this class falls water, similarly used or used in a vacuum or ejector system.

Class (3) Refrigerants of low boiling point, such as ethane, including possibly atmospheric constituents such as oxygen, not used in conventional refrigerating cycles.

Class (4) Various substitute refrigerants developed to overcome the traditional disadvantages of the conventional refrigerants. Prominent among these are several fluorine compounds, led by "Freon-12" having vapor pressure relations that resemble ammonia.

Comparison:—In Table I will be found standard ton comparative data of six (6) principal refrigerants. The values for the standard ton conditions (200 B.T.U. per minute) are based on the A.S.R.E. rating in which the following conditions are considered standard:

(1) Evaporating temperature of plus (+) 5 degrees F.
(2) Condensing temperature of 86 degrees F.
(3) Liquid before expansion valve at 77 degrees F.
(4) Suction gas temperature at 14 degrees F.
(5) Adiabatic compression with constant entropy.

Characteristics:—The characteristics of five (5) of these principal refrigerants are graphically shown in Figure I. For convenience sake, these characteristic curves are based on 86 degrees F. discharge temperature and + 5 degrees F. suction temperature without providing for the 9 degrees F. superheating of the gas before compression and the 9 degrees F. subcooling of the condensate. For each refrigerant cycle the curve at the left denotes the liquid line, the middle curve—the saturated line and the right curve—the compression line at constant entropy. The lower broken line of each refrigerant cycle denotes the evaporation stage, the upper broken line the condensation stage based on + 5 degrees F. and 86 degrees F., respectively, and the vertical broken line denotes the expansion stage.

The following comments on the various refrigerants are of interest:

Ammonia (NH_3):—Ammonia is tolerable in the very small portion of .3 parts per thousand. Local injury in the way of skin eruption, violent respiratory reflexes, inflammation of the eyes and

TABLE 1.

CHARACTERISTICS OF REFRIGERANTS.

(BASED ON A. S. R. E. RATING.)

	(Freon-12) CCl_2F_2	(Carbon Dioxide) CO_2	(Ammonia) NH_3	(Sulphur Dioxide) SO_2	(Methyl Chloride) CH_3Cl	(Carrene) CH_2Cl_2
Melting point—degrees F.	—247.0	—160.6	—107.9	—98.9	—143.7	—142.0
Boiling point—degrees F.	—21.7	—108.4	—28.0	14.0	—10.6	103.7
Critical temperature—degrees F.	233.0	87.8	271.2	314.8	289.6	421.0
Critical pressure—pounds absolute	580.0	1071.0	1651.0	1141.5	969.2	670.0
Suction temperature—degrees F. (including 9 degrees superheat)	14.0	14.0	14.0	14.0	14.0	14.0
Discharge temperature—degrees F.	86.0	86.0	86.0	86.0	86.0	86.0
Liquid temperature—degrees F.	77.0	77.0	77.0	77.0	77.0	77.0
Suction pressure—pounds absolute	26.51	331.9	34.27	11.81	20.9	2.5″ Hg.
Discharge pressure—pounds absolute	107.9	104.3	169.2	66.5	96.0	21.6″ Hg.
Heat of liquid—B.T.U. per pound	25.56	70.50	128.50	39.01	43.30	26.2
Heat of vapor—B.T.U. per pound	80.04	142.35	618.70	185.17	197.40	167.2
Entropy	.17317	1.3085	1.3368	.4002	.4258	.3618
Specific gravity at plus (+) 5 degrees F.	1.445	.985	.660	1.475	.980	1.336
Temperature at compression—degrees F.	106.5	176.1	222.7	202.9	165.5	....
Superheat at discharge—degrees F.	20.5	90.1	136.7	116.9	79.5	....
Refrigerating effect—B.T.U. per pound	54.5	71.9	490.0	146.2	154.1	141.0
Specific volume at suction—cubic feet per pound	1.516	.293	9.224	6.880	4.656	48.31
Pounds of refrigerant per ton per minute	3.671	2.783	0.408	1.370	1.298	1.420
Cubic inches per ton per minute	9618	1409	6502	16280	10440	118,678
Refrigeration per cubic foot—B.T.U.	36.0	245.0	53.0	21.0	33.0	2.9
Compression Ratio	4.07	3.14	4.93	5.63	4.58	8.50

sharp headaches are known to be a result of exposure. In the case of very low temperatures, the large specific volume of NH_3, thus determining the size of the machinery, is a detriment when compared to CO_2. Ammonia is considered as slightly inflammable. However, practically speaking, ammonia remains the standard industrial, as well as commercial, refrigerant in view of its high refrigerating effect and relatively low pressure required. Copper-bearing metals should be avoided when using NH_3.

Carbon Dioxide (CO_2):—Carbon dioxide in concentrations of 2 per cent reduces human efficiency. It is dangerous in concentrations of 4 per cent if the oxygen content is reduced to 13 per cent. In ventilation, 1 part in 1000 has been considered the maximum in conditioned air. A 10 per cent concentration cannot be tolerated over a minute. Excessive pressures to obtain desired temperatures greatly handicap the use of CO_2 no matter how desirable its other properties. Furthermore, these high pressures tend to force more liquid through the regulating valves than is required. This refrigerant finds special uses *only*, on account of its requiring more costly machinery and the continued expense of replenishing escaped gas. Brass, bronze and copper may be used with CO_2.

Methyl Chloride (CH_3Cl):—Methyl chloride is regarded as having relatively small hazards, though it will act as an anaesthetic when concentrated in closed spaces or on long exposures. Methyl chloride vapor in concentrations of from 2 to 2½ per cent by volume (2.62 to 3.28 pounds per 1000 cubic feet) in air kills or seriously injures in 2 hours, as found by the Underwriters' Laboratories. The U. S. Public Health Service Bulletin No. 185 gives a figure for methyl chloride of 0.5 to .1 per cent by volume (1 to 2 ounces per 1000 cubic feet) as the maximum amount for a prolonged exposure to produce a physiological response. In all tests conducted, it has been noted that there is a definitely delayed toxic action and since it is not unpleasant and but faintly noticeable in small quantities, it no doubt presents a greater hazard than some of the highly irritating refrigerants, such as, sulphur dioxide and ammonia, which possesses distinct warning properties. Methyl chloride is moderately flammable, being much less hazardous than gasoline, but more hazardous than ammonia. In the past, this gas has been used extensively in the small commercial field.

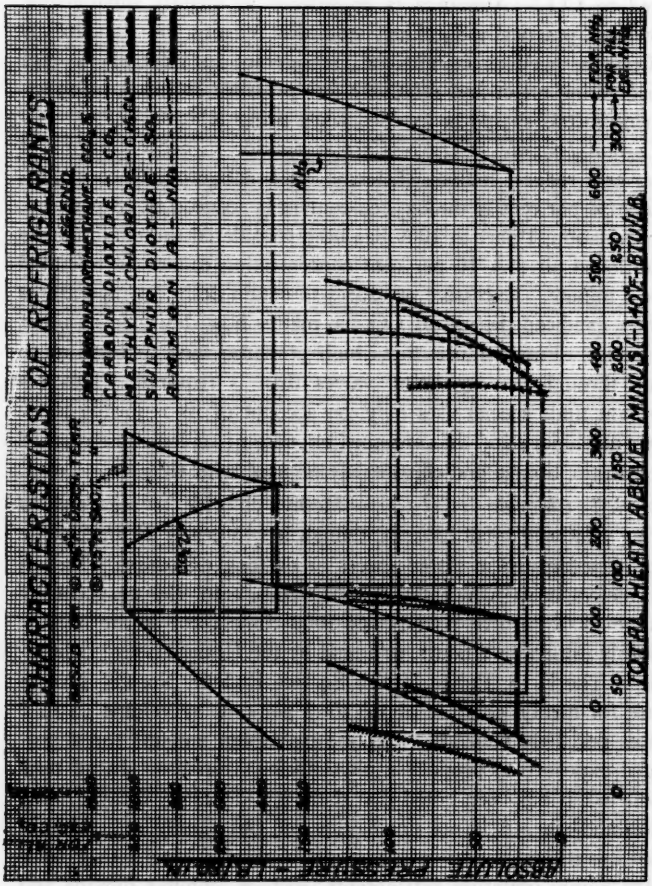

CHARACTERISTICS OF REFRIGERANTS
AMMONIA
CARBON DIOXIDE - CO2
METHYL CHLORIDE - CH3Cl
SULPHUR DIOXIDE - SO2
AIR
ABSOLUTE PRESSURE - LBS. PER SQ. IN.
TOTAL HEAT ABOVE MINUS (-) 40° F. - B.T.U./LB.

FIGURE I.—CHARACTERISTICS OF REFRIGERANTS.

Sulphur Dioxide (SO_2):—Sulphur dioxide is detectable in very small concentrations, 3 p.p.m., while at 500 p.p.m. it is suffocating. This gas does not act as a poison. Sulphur dioxide operates at a lower pressure than methyl chloride or "Freon-12" to obtain desired temperatures which some engineers consider to be an advantage, but it is to be pointed out that SO_2 machines will operate at negative pressures on the low side. This condition will cause the infiltration of moisture-laden air. Sulphur dioxide is much more toxic than methyl chloride or "Freon-12." It has been widely used in small household refrigerators where small charges are involved but the tendency at the present time is to use refrigerants that are less hazardous. Brass, bronze and copper may be used with SO_2, but all systems must be thoroughly dried to eliminate the formation of sulphurous acid.

Methylene Chloride (Carrene—CH_2Cl_2):—"Carrene" is a non-explosive, non-flammable, non-corrosive anaesthetic compound of carbon, hydrogen and chlorine having a slight, pleasant odor. Its chief merit is that owing to its low vapor pressure, leaks of refrigerant will normally not escape. The refrigerant is to some extent toxic, much more so than "Freon-12," and slightly hazardous to the health and life of Navy personnel. "Freon-12" in concentrations as high as 20 per cent by volume (63 pounds per 1000 cubic feet) is non-toxic for exposures of two hours. This refrigerant has been used in centrifugal type compressors especially for air conditioning in industrial plants, but is now being supplanted with "Freon-11."

Other refrigerants not used in any great degree are *ethane, butane, propane, ethyl-bromide, dichloroethylene, methyl-bromide, methyl-formate* and *ethyl-chloride.*

Packing containing rubber should not be used with the hydrocarbons, ethyl chloride or methyl chloride. Gaskets of lead, fiber or asbestos are recommended.

The characteristics and properties of "Freon-12," which is now being used on all Naval vessels under construction, will be handled as a separate subject.

"FREON-12" AS A REFRIGERANT.

Physical Properties:—Dichlorodifluoromethane is a halofluoro derivative of methane marketed under the trade name of "Freon-12." It resembles carbon tetrachloride, the fire extinguishing fluid, in odor and at ordinary temperatures it is a liquid when under a pressure of about 75 pounds. The gas is shipped liquefied under pressure in steel cylinders, meeting the specifications of the Interstate Commerce Commission, similar to those used for low pressure compressed gases.

The color of "Freon-12" is water-white and free from cloudiness. It is odorless in concentrations of less than 20 per cent by volume in air, which is equivalent to 63 pounds of liquid vaporized into a space of 1000 cubic feet. Its vapor or vapor-air mixture, higher than 20 per cent, is mildly ethereal. The boiling point, after initial temperature equilibrium has been attained, shall not be higher than (—) 20.2 degrees F. at a barometric pressure of 29.92 inches mercury (superheating of the liquid causes observed boiling point to be consistently 1.2 degrees F. higher than the true boiling point of the sample), and the boiling range from the initial boiling point, until 15 per cent by volume of the sample remains, should not exceed .9 degrees F.

Manufacture:—The following is quoted from a paper "Organic Fluorides as Refrigerants" written by Thomas Midgeley, Jr., and Albert L. Henne*:

"The reaction involved in the manufacture of "Freon-12" is as follows:

$$3CCl_4 + 2SbF_3 \xrightarrow[SbCl_5]{} 3CCl_2F_2 + 2SbCl_3$$

"When pure and dry antimony trifluoride (obtained by sublimation of the crude material) is brought into contact with carbon tetrachloride, in the presence of a small amount of antimony pentachloride, fluorine substitutes the chlorine with great ease in the carbon tetrachloride."

Decomposition:—All refrigerants, such as methyl-chloride, "Carrene," carbon dioxide, sulphur dioxide and "Freon-12," in

* Published in Industrial and Engineering Chemistry, Vol. 22, p. 542, May, 1930.

common with all organic materials, decompose on passage through a naked flame of high temperature. But it is considered very unlikely that all of the conditions required to cause decomposition of the refrigerant will occur simultaneously, except under laboratory test conditions and certainly not under any habitable condition.

The only hazard which may be introduced through the use of "Freon-12" as a refrigerant is a very remote health hazard which might be presented should leakage occur and the vapors come into direct contact with an open flame of high temperature and be decomposed. Even in such a case, in order to bring about an appreciable health hazard, there must be the following combination of circumstances:

(*a*) The release of a large quantity of "Freon-12" in a tightly confined space.
(*b*) The presence of flame or other high temperature.
(*c*) Circulation of the atmosphere within the space so that the heavy vapors of the refrigerant will be brought into direct contact with the flame or high temperature, for only those vapors in direct contact will be decomposed.
(*d*) No ventilation to the outside such as would be provided by open doors or mechanical ventilation, for if there is ventilation to the outside, the vapors of the refrigerant and whatever products of decomposition there are will escape.

Finally, if these circumstances were all present, the harmful products of decomposition would be pungent and irritating, rendering them noticeable in minute quantities and giving ample warning to prevent prolonged exposure long before concentrations dangerous to health are reached. The decomposition products are readily dissolved by water from hose streams.

The odor or irritating properties of decomposition are not to be confused with harmful concentrations, also the odor or irritating properties of combustion from the flame are not to be confused with decomposition products. This subject of decomposition applies not only to "Freon-12," but to all refrigerants and such

other materials as: paper, paint, wood, gasoline, rubber insulation, wool, etc. Gases from different materials differ greatly in toxicity with materials containing nitrogen or sulphur producing the most toxic gases.

In conclusion, it may be stated that " Freon-12 " is

(1) A non-combustible, odorless, non-irritating, non-explosive, non-inflammable refrigerant, and will not cause a panic or fire hazard. It is as non-inflammable as carbon tetrachloride.
(2) A stable compound capable of undergoing without decomposition the physical changes to which it is commonly subjected in service.
(3) It does not corrode iron, steel, copper, brass, Monel metal, etc., and
(4) Is practically a non-toxic gas. It is less toxic than carbon dioxide and a very satisfactory refrigerant.

Design Factors:—The volume per pound of the refrigerant vapor, together with the refrigerating effect per pound, determines the piston displacement. The actual piston displacement is dependent on the evaporator pressure or temperature, capacity of the compressor, compressor speed, superheating effect of the cylinder walls, refinement in design, compression clearance, design of valves, etc. It will be noted from Table I that the theoretical piston displacement of 9618 cubic inches per minute per standard ton for " Freon-12 " charged compressors exceeds that required for CO_2 and NH_3, but is smaller than that required for the remaining refrigerants. " Freon-12 " has a relatively low refrigerating effect, thereby requiring more liquid refrigerant in circulation to produce the desired amount of refrigeration. This represents an advantage in that the circulation of larger quantities of liquid refrigerant will permit the use of less sensitive, more accurate and more positive operating and regulating mechanisms.

Moisture:—" Freon-12 " compressors operate at a low but positive back pressure which permits high volumetric efficiency besides preventing moisture-laden air from accidentally entering the sys-

tem, and also permits the testing for leaks. The presence of moisture in the system causes the following adverse conditions:

(1) Reduction of performance by air lodging in the condenser, thus causing high head pressures and temperatures,

(2) Moisture freezing out as ice and interfering with the operation of the expansion valves,

(3) Moisture causing intercrystalline embrittlement of sylphons, bellows, diaphragms, etc., when under cyclic stress.

Velocities:—The limits in design of a refrigerating plant of the "Freon-12" vapor or gas velocities in feet per minute are given in the following table. The limiting velocities of "Freon-12" liquid are between 80 and 100 feet per minute.

	Large pipe sizes	*Small tubing sizes*
Suction line	2750 to 3750	800 to 1800
Suction valve	2500 to 3500	750 to 1750
Discharge valve	4000 to 5000	2000 to 2500
Discharge line	3750 to 4750	1750 to 2250

Lubrication:—"Freon-12" is completely miscible with mineral lubricating oil; therefore, an oil of high viscosity is required. The kind (Navy Contract Oil) of oil also depends on the type, size and speed of the compressor among other factors. The oil should be straight run, properly refined mineral oil and must be free from all impurities (dehydrated) especially water. In a direct expansion system as employed aboard ship, there will be no oil logging of the evaporator due to the high velocity of the refrigerant vapor or gas in the return line pushing along any oil particles. The amount of oil absorbed by "Freon-12" depends on the oil characteristics, temperature of the oil in the crankcase and the operating back pressure in the compressor crankcase.

Figure II shows the characteristics of "Freon-12-oil" solutions. In the upper right hand quadrant is shown the per cent by weight of "Freon-12" absorbed in 150 and 325 viscosity (at 100 degrees F.) mineral oils at various back pressures and oil temperatures. More "Freon-12" is absorbed by the oil at higher pressures and at lower temperatures. Lighter oils absorb less "Freon-12" and

heavier oils absorb more "Freon-12." During off cycles the oil temperature is lowered, and the back pressure is increased, thus permitting the "Freon-12" vapors to become more absorbed by and condensed in the mineral oil. At the time the compressor starts operation, the back pressure in the low side of the crankcase is suddenly reduced, with the result that the "Freon-12" in solution with the crankcase oil will boil, causing the lubricating oil to foam.

In the upper left hand quadrant is shown the vapor pressure and boiling point of "Freon-12-oil" solutions with the solutions containing varying percentages of oil by weight. In the lower two quadrants are shown the viscosity of "Freon-12-oil" solutions on the basis of 150 and 325 viscosity (at 100 degrees F.) mineral oils.

In the reciprocating compressor, where there is the possibility of some of the oil passing over to the high pressure side to become mixed with the "Freon-12" there is provision for return directly to the crankcase. The preferred type of compressor is one in which the refrigerant vapors are kept entirely apart from the crankcase, thereby largely eliminating the possibility of mixture with oil at least at this point. This enables the oil to maintain its original viscosity or merely to follow the normal reduction in viscosity, which would take place as the crankcase comes up to average operating temperature. This condition will prevail in the enclosed crankcase machine, equipped with trunk-type pistons and designed for pressure lubrication. The oil pump maintains positive oil circulation without excessive splash effect; therefore, foaming is markedly decreased. Reduction of oil splash in turn reduces the tendency of any refrigerant present to mix with the oil supply, especially as there is no circulation of refrigerant vapors within the crankcase. An oil of somewhat higher original viscosity must be used than would be necessary if the refrigerant were non-miscible with the oil. The conditions which determine the proper kind of oil are speed, intake and discharge temperatures, means of cooling and the method of application of the oil. For the type of compressors installed aboard Naval vessels an oil of from 150 to 300 S.S.U. viscosity at 100 degrees F. is recommended. The normal content of "Freon-12" in oil varies from 10 to 12 per cent in average service.

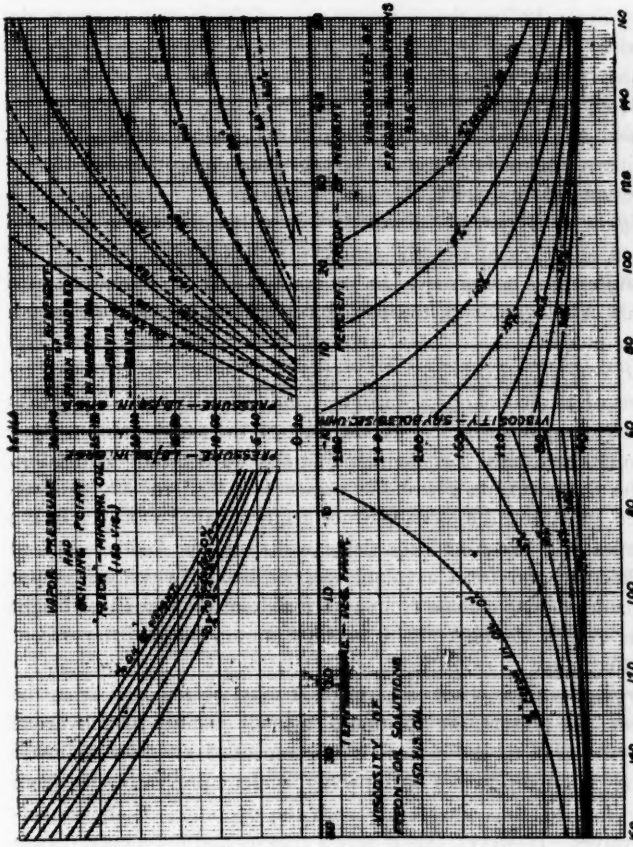

OIL TEMPERATURE – DEG. FAHR.
OIL TEMPERATURE – DEG. FAHR.
VISCOSITY – SAYBOLT SEC. UNIVERSAL
PRESSURE – LBS. PER SQ. IN. GAUGE
PRESSURE – LBS. PER SQ. IN. GAUGE
PERCENT BY WEIGHT
TEMPERATURE – DEG. FAHR.
SOLUBILITY OF FREON-OIL SOLUTIONS IN FREON
MIXED PRESSURE AND DENSITY FREON-OIL SOLUTIONS (FREON-ARMOUR OIL 140-150)

FIGURE II.—"FREON-12"—OIL SOLUTIONS.

The oil is partly separated from the " Freon-12 " vapor by means of a surge drum installed in the suction line. The surge drum contains a baffle to throw down oil particles and thereby acts in similar fashion to an oil separator. The separation of oil from the " Freon-12 " in the discharge side of the system is practically impossible due to the high pressures, temperatures and velocities of the refrigerant.

Filters:—Since " Freon-12 " has the ability of loosening and removing any and all particles of dirt and scale, then gas filters or strainers employing fine mesh screens must be installed in the refrigerant piping system.

Effect on Metals:—" Freon-12 " in either the vapor or liquid stages, and even at elevated temperatures under the most adverse conditions, has no corrosive action upon any of the commonly used metals or alloys nor on the tensile strength or per cent elongation of the metals.

" Freon-12" Leakage:—" Freon-12 " leakage is detected by means of an alcohol or gas burning halide lamp or torch fitted with a rubber exploring hose. In the case of the pump-pressure type of torch, care must be taken so that the torch is not pumped-up in the compartment where the presence of " Freon-12 " is suspected, otherwise the torch will not perform satisfactorily. In the presence of " Freon-12 " the flame will cause the " Freon-12 " to break down and form a volatile copper halide, which will cause the flame color to change from a normal colorless flame to a bright green, even if the air contains as small an amount as .01 per cent of " Freon-12."

Drying:—Since water is harmful, it must be removed from the system. At present, the addition of a small quantity of anhydrous methyl alcohol to the system is recommended in order to lower the freezing temperature of the water, thereby preventing ice from being formed, thus interfering with the operation of the expansion valves. However, this is only a remedy and not a cure. Some manufacturers propose the use of silica-gel or activated alumina in the form of cartridges for the purpose of drying the " Freon-12 " charge. The water can also be removed by the circulation of dehydrated air or by evacuation and by heating the units to about 180

degrees—200 degrees F. The use of calcium chloride is not recommended as it is possible that if an excessive amount of moisture is present, a solution of calcium chloride would be circulated through the system and then corrosion may result.

Miscellaneous:—Special precautions must be taken to properly evacuate, charge, purge and to add oil to a "Freon-12" system. The instructions for each make of compressor unit are slightly different due to differences in design and construction. The charging of a "Freon-12" system is best accomplished by the use of a "Kerotest" charging stand, wherein the system is charged with the proper amount of "Freon-12" from a 113½-pound standard supply cylinder by means of pulling a vacuum of about 28 to 29 inches of mercury on the compressor. In adding oil to the system extreme care must be taken to prevent air from being drawn in while the oil is being added. The oil is likewise added to the system under a vacuum condition in the crankcase.

PRINCIPLES OF REFRIGERATION.

Definition of Ton of Refrigeration:—The basic unit of measurement of the cooling action of a refrigerant machine is the ton, equal to 12,000 B.T.U. per hour. The unit of refrigeration as a quantity is the ton-day equal to 288,000 B.T.U. per day. This quantity of heat is approximately equal to the latent heat required in freezing one (1) ton of ice per day, *i. e.* $2000 \times 144 = 288{,}000$ where 144 represents the latent heat of fusion.

The rated capacity of a machine on the basis of the A.S.R.E. (American Society of Refrigerating Engineers) tonnage rating is measured in tons referred to the standard conditions previously outlined. However, the standard Navy tonnage rating for refrigerating machines is based on minus (—) 5 degrees F. evaporating temperature with no super-heat of the gas before entering the compressor.

Process of Refrigeration:—The process of refrigeration is most commonly accomplished by the evaporation of a liquid refrigerant, thereby extracting heat from the medium to be cooled. The refrigeration cycle is then composed chiefly of three (3) further steps, whose purpose is to remove this same heat from the evaporating refrigerant by again putting it in the liquid state in order that it

may be used repeatedly in a continuous process. To do this, it is necessary that the gas be compressed, thereby permitting condensation at the normal temperatures of available water or air. This is generally done by compressors requiring mechanical energy (work).

In other words, to complete the cycle the following changes take place so far as Navy refrigerating or air conditioning installations are concerned:

(1) Compression of gas—Mechanical driven compressor, either reciprocating or rotary,
(2) Condensation of gas—Either by water or air cooled unit,
(3) Expansion of liquid—Either by thermal or pressure actuated expansion valves.
(4) Evaporation of liquid—Either by

A. Coils in cold storage compartments.
B. Air conditioning unit.
C. Process water heat exchanger.
D. Ship's service store equipment.
E. Circulating water (under reverse cycle operation).

Various Refrigerating Systems:—The different systems of mechanical referigeration can be divided into three main groups: (1) air systems, (2) vapor-compression and (3) vapor-absorption systems.

In the first group (air systems), the closed cycle represents the principle on which Allen dense air machines operate. The two main reasons why these cold air machines were superseded by the vapor-compression machines are: (1) very low actual coefficient of performance (not over .75), and (2) high initial cost of installation. This paper will only discuss the vapor-compression system.

Definition of Terms:—The refrigerating effect is the amount of heat which can be absorbed from a system by one (1) pound of the refrigerant under given conditions of temperature and pressure. The coefficient of performance is the ratio of the refrigerating effect to the work of compression. The coefficient of performance for "Freon-12" systems is about 4.7 as compared with .75 for the Allen dense air machine.

Parts of Refrigerating Machine:—Figure III shows a diagram of a refrigeration cycle as applied to cold storage refrigeration, air cooling and air heating (reverse cycle) systems. The refrigerating machinery with which the heat is removed from the air or water consists of three (3) essential parts—the evaporator, the compressor and the condenser besides expansion valves, surge drums, piping, oil traps, motors, control equipment, etc.

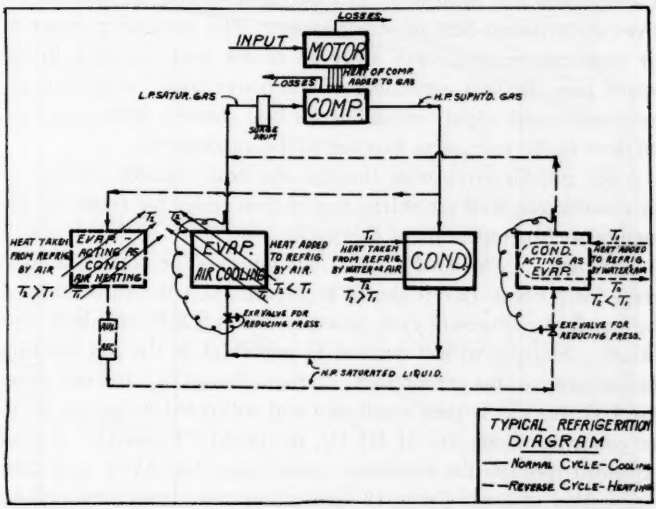


FIGURE III.—DIAGRAM OF REFRIGERATION CYCLES.

Evaporator:—In the evaporator or cooler a pressure is maintained which causes the liquid refrigerant to evaporate or boil at a sufficiently low temperature to absorb heat from the water, air or other medium in contact with the cooling surface. Various devices such as separators, accumulators and eliminators are required to remove any entrainment of liquid from the gas leaving the cooler.

Compressor:—The function of the compressor is to withdraw the gaseous refrigerant from the cooler and deliver it to the condenser at such a pressure that its heat can be absorbed by air or water at ordinary temperatures. At this point, the discharge control serves to divide the low and high pressure sides of the system

and maintain maximum efficiency of operation by preventing leakage of the compressed vapor back into the low pressure side.

Condenser:—The condenser whether air cooled or water cooled serves to condense the compressed gaseous refrigerant. It is then capable of serving as a cooling medium, and this is brought about by passing it through an expansion or regulating valve on the expansion side of the system. Here, by virtue of a drop in pressure, it evaporates and returns to its gaseous state and, in so doing, it gives up its latent heat of vaporization. The circulating water or air must receive and carry away all of the heat absorbed in the cooler, plus the heat equivalent of the power used for driving the compressor with slight corrections for heat gains or losses from the surfaces of the equipment exposed to the atmosphere.

After suitable circulation through the heat transfer element or air conditioning unit the refrigerant is then ready for return to the compressor for repetition of this cycle.

"Freon-12" Standard Cycle Conditions:—The total heat diagram for "Freon-12" is shown in Figure IV, wherein is outlined the standard refrigerant cycle based on the A.S.R.E. standard conditions. Starting with 9 degrees F. superheat in the gas entering the compressor, the compression of the "Freon-12" follows along line I-II, the gas is then condensed and subcooled 9 degrees F. in the condenser along line II-III-IV, the liquid "Freon-12" is then expanded through the expansion valve along line IV-V and then evaporation of the "Freon-12" absorbing heat from surrounding air or water takes place along the line V-I.

Graphical Characteristics of "Freon-12":—Figure V graphically illustrates the characteristics of "Freon-12" as a refrigerant. The refrigerating effect in tons per minute for varying discharge pressures and evaporating (suction) temperatures is plotted for both A.S.R.E. and Navy standard tonnage ratings. In addition,

(1) The theoretical volume of gas required per ton per minute,
(2) The horsepower of compression per ton,
(3) The heat removed by the condenser water per minute per ton, and
(4) The refrigerating effect per pound of "Freon-12" in B.T.U.

are plotted for varying discharge pressures and evaporating (suc-

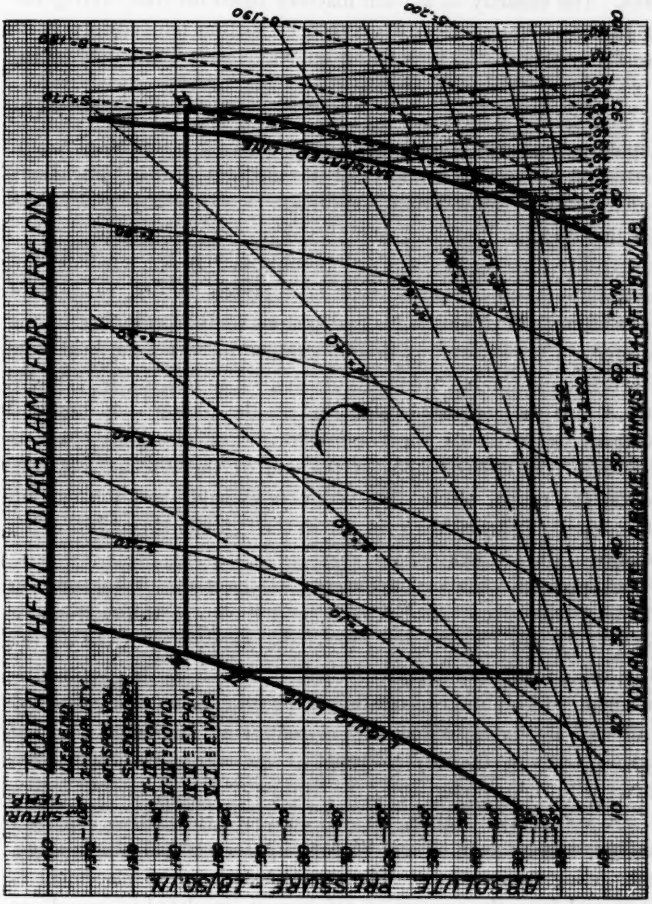


FIGURE IV.—TOTAL HEAT DIAGRAM FOR "FREON-12."

tion) temperatures corresponding to the suction pressures. As noted from these graphs, the capacity of a unit is reduced with lower evaporating temperatures and with higher discharge pressures. The capacity of a 1-ton machine based on Navy refrigerating rating of minus (—) 5 degrees F. suction temperature is increased to almost 2½ tons based on Navy air conditioning rating of 35 degrees F. suction temperature.

Compressor Performance:—The reciprocating compressor is best adapted to high operating heads, the centrifugal compressor to low heads and the rotary compressor to medium heads. For small capacities, the reciprocating compressor has inherent advantages over the others in first cost, flexibility and cost of operation.

In reality, the compression in a compressor is not adiabatic but polytropic—in other words, heat is absorbed by the refrigerant vapor from the cylinder walls and pistons at the beginning of the compression stroke, and as its temperature increases the heat flow is reversed and the hot vapor begins to give up heat to the cylinder walls. This cooling of the vapor lowers the compression line and decreases the work of the compressor. The clearance space is filled with compressed vapor at the end of the discharge stroke, and on the return stroke this vapor re-expands and the suction valves open only when the pressure in the cylinder drops below the pressure in the evaporator.

The apparent volumetric efficiency is the ratio of the suction volume to the piston displacement, and the total volumetric efficiency is the ratio of the capacity of the compressor based on the actual amount of vapor taken in and delivered to the piston displacement. In addition to clearance, the total volumetric efficiency is affected by the resistance and wiredrawing in the suction and discharge valves, the superheating of the vapor resulting from contact with the cylinder walls, the heating of the piston and valves by hot compressed vapor, etc. The total volumetric efficiency usually varies between 65 and 85 per cent for vertical, single stage, single acting compressors. The theoretical piston displacement is determined from the amount of refrigerant circulated in pound per minute and the specific volume at suction in cubic feet per pound. The actual piston displacement must be larger than the theoretical

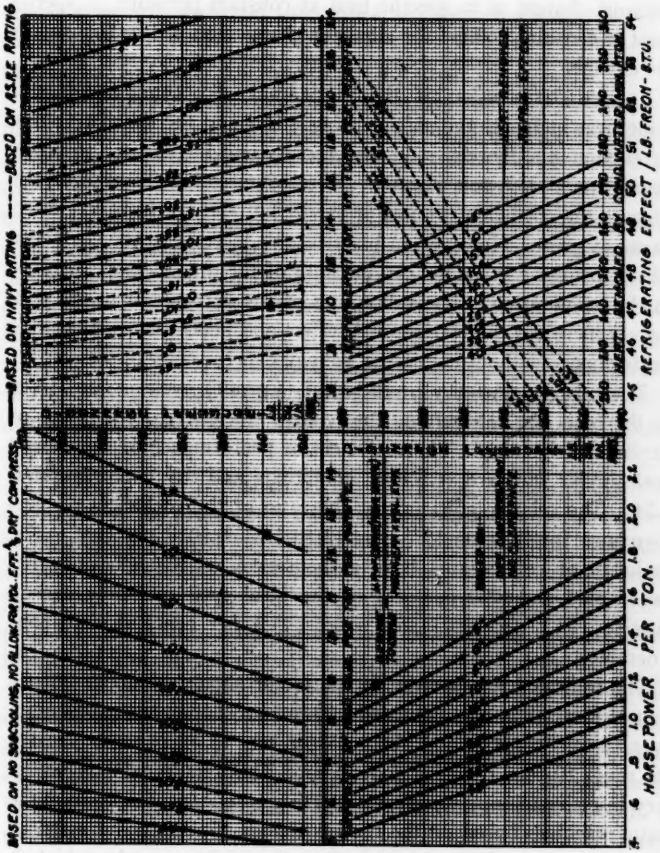

BASED ON NO SUBCOOLING, NO ALLOWANCE FOR VOL. EFF. & PIPE COMPRESS. ——BASED ON NAVY RATING ----BASED ON A.S.R.E. RATING
HORSE POWER PER TON.
REFRIGERATING EFFECT / LB. FREON-12 B.T.U.

FIGURE V.—"FREON-12" CHARACTERISTICS.

value because of clearance, superheating effect, valve resistance, etc. In other words, actual p.d. = theoretical p.d. ÷ total volumetric efficiency.

In a polytropic change, pv^n = constant—where n = 1 for isothermal change, n = specific heat at constant pressure ÷ specific heat at constant volume for adiabatic change and n = o for isopiestic change.

In the compression of the refrigerant, dry compression with superheat increases the refrigerating effect and the specific volume of suction gas, thereby reducing capacity and increasing the power requirements per unit capacity. Wet compression theoretically increases the coefficient of performance, but the refrigerating effect is not as great. However, in practice, this increase in the coefficient of performance is not the case because of the time element involved in the evaporation. The subcooling of the refrigerant in the condenser increases the refrigerating effect and hence the coefficient of performance without increasing the work of the compressor.

The relative efficiency of the ideal (Carnot) cycle is considered as the basis for determining the relative efficiency of the cycles of various refrigerants. The coefficient of performance of the Carnot cycle under standard A.S.R.E. conditions and dry compression is 5.74. The relative efficiency or the ratio of the coefficient of performance for "Freon-12" of 4.72 to 5.74 is 82.3 per cent. The majority of refrigerants have a theoretical relative efficiency between 89.4 and 82.0 per cent of the ideal cycle and only carbon dioxide (44.7 per cent) and water (71.5 per cent) deviate considerably.

The indicated horsepower of the motor should be about 25 per cent greater than the indicated compressor horsepower for units over 10 tons capacity and about 35 per cent to 50 per cent greater for units 10 tons capacity and under. This requirement is necessary in order to supply sufficient starting torque, especially in automatic machines based on the horsepower of the gas compressed at the highest probable condenser and suction pressure under which it may operate.

A large compression ratio is undesirable in reciprocating compressors from the standpoint of clearance losses and may make the use of compound compression necessary.

Condenser Performance:—Condensers are generally divided into three (3) types, namely—double pipe, shell and tube, and atmospheric. The shell and tube condenser is the only type used in "Freon-12" cold storage refrigeration installations, while both double pipe and shell and tube types are used in carbon dioxide installations. However, for air conditioning installations shell and tube and atmospheric type condensers are used.

In the shell and tube condenser, the refrigerant is condensed in the shell and the circulating water is distributed through the tubes. In the atmospheric condenser the refrigerant is condensed in the tubes, usually of finned construction, by the circulation of air across the finned tubes. The shell of the condenser is usually large enough so that it also acts as a receiver. Dependent upon varying circulating water velocities and condition of the tubes, the heat transfer coefficient of condensing "Freon-12" usually varies between 50 and 90 B.T.U. per hour per square foot per degree M.T.D. In air cooled condensers this heat transfer coefficient varies under similar conditions between 8 and 14. It is important that the condenser be large enough in order to prevent the building up of high discharge pressures.

If there is poor circulation of the condensing media through the condenser or the presence of oil or dirt on the surface of the tubes or high operating back pressures together with superheat from the compressor, then the temperature of the refrigerant will be from 15 degrees F. to 40 degrees F. above the circulating water temperature. Operating head pressures, in general, will vary with the temperature of the circulating water.

Evaporator or Cooler Performance:—The performance of the evaporators or coolers which take the place of coils in the cold storage compartments, air conditioning units, process water heat exchangers, ship's service store equipment, etc., will be discussed in Part II of this paper, which will treat with each installation in a practical manner. The type of compressor and condenser is practically the same irregardless of the application of the refrigerant cycle and therefore these units were discussed in Part I of this paper.

Refrigeration as Applied to Air Conditioning:—Refrigeration is employed to maintain low temperatures and/or relative humidities

for the improvement of comfort in various rooms and for the maintenance of low temperatures in various compartments aboard ship.

The refrigerating equipment as applied to air conditioning is identical with that for cold storage refrigeration, except that the operating suction temperature is maintained at (+) 35 degrees F., thereby increasing the capacity of the machine about 2½ times. The evaporator for an air conditioning plant is usually of finned tube construction with the air forced over the surface by a motor-driven fan.

Reverse Cycle:—In submarine installations, first use has been made of operating the air conditioning plant under reverse cycle conditions wherein the air is heated in the winter time. Under these conditions, the finned tube evaporator acts as the "Freon-12" condenser wherein heat is imparted to the air by condensing "Freon-12." The condenser (shell and tube unit) then acts as the evaporator wherein the "Freon-12" is evaporated, thereby extracting heat from the circulating water. The disadvantage of reverse cycle operation is that with the falling off of the compressor load, the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which in turn would fracture, thus allowing the refrigerant to escape in the water. For this reason, a low temperature water failure switch should be employed that will shut down the compressor motor when the circulating water temperature reaches about 34 degrees F.

The operation of the reverse cycle should be limited to sea water temperatures of from 40 degrees to 50 degrees F. It is possible to operate with sea water temperatures lower than 40 degrees F. but as sea water approaches the refrigerant temperature, the capacity of the unit drops and, on the other hand, temperatures higher than 50 degrees F. are likely to build up sufficient pressure on the suction side to overload the motor which can be corrected by the installation of a suction pressure regulating valve. In the piping arrangement, an auxiliary receiver is necessary on the heating cycle since there is very little storage space for the refrigerant in the coils exposed to the air stream. The condensed refrigerant filling up these coils would produce an exceptionally high discharge pressure. Under reverse cycle conditions, the oil level in the crankcase of the compressor must be watched very carefully.

TABLE II.

FORMULAE FOR CALCULATING PERFORMANCE OF "FREON-12" REFRIGERATING PLANTS.

No.	Item	Formulae	Values for A. S. R. E. Conditions.	Values for Navy Conditions.
1	Compression ratio	Discharge pressure ÷ suction pressure	4.07	5.13
2	Refrigerating effect, B.T.U. per pound	Heat at suction — heat of liquid	54.5	49.8
3	Heat of compress., B.T.U. per pound	Heat at discharge — heat at suction	11.5	12.3
4	Heat rem. by condenser, B.T.U. per lb.	Heat at discharge — heat of liquid	66.0	62.1
5	Quan. of refrigerant, lb. per ton per min.	200 ÷ refrigerating effect (item 2)	3.671	4.02
6	Vol. of ref., cu. in. per ton per minute	Item 5 × specific volume at suction	9618	12530
7	Theoretical horsepower per ton per min.	Item 3 × item 5 ÷ 42.42 (1 horsepower = 42.42 B.T.U. per minute)	1.00	1.17
8	Coefficient of perf. (Carnot = 5.74)	Item 2 ÷ item 3	4.72	4.05
9	Cycle efficiency, per cent	Item 8 ÷ 5.74	82.3	70.5
10	Heat rem. by condenser, B.T.U. per ton per minute	Item 4 × item 5	242.0	250.0
11	Condenser water, pound per ton per min.	Item 10 ÷ t. r. (assume 6 degrees F.)	40.4	41.6
12	Refrigeration, B.T.U. per cubic foot	200 ÷ (item 5 × specific volume at suction)	36.0	27.6
13	Act. piston disp., cu. in. per ton per min.	Item 6 ÷ volumetric efficiency (assume 63 per cent)	15270	19900
14	App. volumetric efficiency, per cent	Volume at suction ÷ piston displacement	....	....
15	Total volumetric efficiency, per cent	Capacity of compressor ÷ piston displacement	....	....
16	Real volumetric efficiency, per cent	Item 15 ÷ item 14	....	....
17	Mechanical efficiency, per cent	Indicated horsepower of compression ÷ brake h.p.	....	....
18	Compressor efficiency, per cent	Hp. required to compress all vapor ÷ ind. hp.	....	....
19	Efficiency of compression, per cent	Item 17 × item 18	....	....
20	Expected horsepower	Item 7 ÷ item 19	....	....
21	Horsepower input to motor	H.P. input to compressor ÷ motor efficiency	....	....
22	Performance Factor	Refrigeration output ÷ Pwr. input to compressor	....	....
23	Overall Factor	Refrigeration output ÷ Pwr. input to motor	....	....

Values for Navy Conditions are based on: (1) — 5 degrees F. suction; (2) 86 degrees F. discharge; (3) no subcooling; (4) no superheat.

Values for A.S.R.E. Conditions are based on: (1) + 5 degrees F. suction; (2) 86 degrees F. discharge; (3) 9 degrees F. subcooling; (4) 9 degrees F. superheat.

In Table II are listed various formulae and terms commonly used in the calculations of the performance of refrigerating units. The following example is given wherein the various factors making up the volumetric losses of a compressor are calculated.

ILLUSTRATION OF FACTORS MAKING UP VOLUMETRIC LOSSES.

Assumptions.

Stroke = $3\frac{1}{2}$ inches.
Bore = $3\frac{1}{4}$ inches.
Speed = 400 R.P.M.
Number of cylinders = 4.
Suction pressure = 21.6 pounds absolute.
Discharge pressure = 107.9 pounds absolute.
Refrigerant condensed = 150 pounds per hour.
Temperature at service valve of compressor = 80 degrees F.
Temperature of vap. bef. comp. stroke = 100 degrees F.
Pressure of vap. bef. comp. stroke = 20.0 pounds absolute.

Calculations.

A—Piston displacement =

$$\frac{3.1416 \times 3.25 \times 3.5 \times 400 \times 60 \times 3.25}{4 \times 1728}$$

= 403 cubic feet per hour per cylinder.

B—Specific volume of saturated "Freon-12" at 21.6 pounds absolute = 1.8, hence:

$$\frac{1.8 \times 150}{403} = 67 \text{ per cent (first volumetric efficiency)}$$

C—Specific volume of "Freon-12" at 21.6 pounds absolute and 80 degrees F. = 2.2, hence:

$$\frac{2.2 \times 150}{403} = 82 \text{ per cent (second volumetric efficiency)}$$

D—If "Freon-12" is all heated to temperature of discharge; specific volume at 21.6 pounds and 100 degrees F. = 2.4, hence:

$$\frac{2.4 \times 150}{403} = 89 \text{ per cent (third volumetric efficiency)}$$

E—Specific volume of "Freon-12" at 20.0 pounds absolute and 100 degrees F. = 2.5, hence:

$\frac{2.5 \times 150}{403}$ = 93 per cent (fourth volumetric efficiency)

F—If effective stroke = $3\frac{3}{8}$ inches then $403 \times \frac{3.375}{3.50} = 389$, hence:

$\frac{2.5 \times 150}{389}$ = $96\frac{1}{2}$ per cent (fifth volumetric efficiency)

Conclusions:—

(1) Reduction in capacity due to leakage of vapor past piston and valves = $3\frac{1}{2}$ per cent (100-$96\frac{1}{2}$).
(2) Reduction in capacity due to clearance volume re-expansion = $3\frac{1}{2}$ per cent ($96\frac{1}{2}$-93).
(3) Reduction in capacity due to wiredrawing in suction stroke = 4 per cent (93-89).
(4) Reduction in capacity due to heating in cylinder before stroke = 7 per cent (89-82).
(5) Loss in capacity because of heating suction vapors = 15 per cent (82-67).
(6) Final volumetric efficiency = 67 per cent.

PART II.

SCOPE.

This part of the paper will deal with the actual refrigerating installations employing "Freon-12" aboard Naval vessels from a practical viewpoint. In general, the installations are of the following types and they will be discussed separately in the order indicated:

Cold storage refrigeration,
Air conditioning and process water cooling,
Ship's service store equipment.

In conclusion, the paper will discuss briefly the subjects of "Freon-12" piping, gaskets, tests and other data that is more or less common to each of the above applications of refrigeration.

COLD STORAGE REFRIGERATION.

Determination of Load:—In determining the size of the refrigerating plant load in tons required for the cold storage compartments, it is necessary to calculate the heat losses through the insula-

tion for all sides of each compartment, taking into account the type, thickness or number of layers of insulation, the resultant coefficient of heat transfer factor in addition to the temperature outside the compartment and the temperature desired in the compartment. In other words, we have the following equation:

Heat loss = surface in square feet × temperature difference × heat transmission factor for insulation.

To the above must be added the estimated losses due to the heat generated within the compartment from lights, fans, opening of doors, men at work, etc., and the amount of refrigeration required to produce the specified amount of ice starting with water at 100 degrees F., freezing the water and then further reducing the ice to a temperature of + 5 degrees F. In addition, the refrigeration of the contents of the room must be taken into account. The amount of refrigeration for this purpose depends on the quantity, temperature and frequency of storage. It is good practice to allow a margin of safety amounting to 50 per cent.

Insulation:—The various kinds of insulation used aboard Naval vessels are cork board, ground cork, fibrous glass or glass wool and aluminum foil. At present, the Navy is experimenting with fibrous glass and aluminum foil in an effort to find the most satisfactory kind of insulation for Naval use. The aluminum foil depends on the reflection of heat from the bright layers of foil, similar to the silvered surfaces of a thermos bottle. The method of application and the number of layers used determine its heat conductivity. Aluminum foil weighs about 1/10 the weight of glass wool and about 1/40 the weight of cork. Therefore, it deserves serious consideration on account of its weight saving features.

Plant Arrangement:—A typical arrangement of refrigerating machinery as installed aboard a Navy cruiser is outlined in Figure VI. The system of control as indicated is by means of a thermostat in the meat room and low pressure control switches in each compressor suction line. In this particular installation the air conditioning of special compartments is accomplished by brine recirculation and use is made of the pressure actuated expansion valve in the liquid line combined with the temperature regulating valve in the suction side of the coils in the compartments.

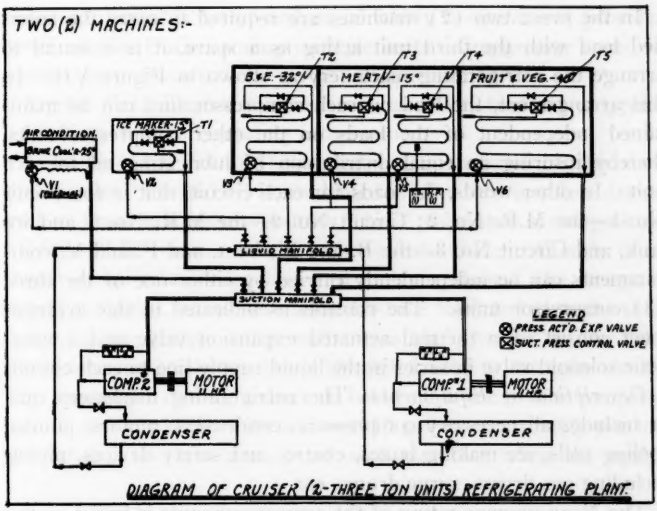


FIGURE VI.—ARRANGEMENT OF TWO MACHINE PLANT.

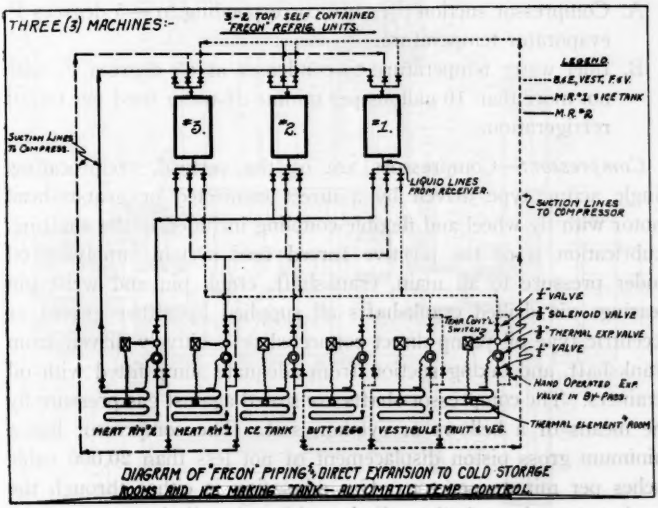


FIGURE VII.—ARRANGEMENT OF THREE MACHINE PLANT.

In the event two (2) machines are required to carry the specified load with the third unit acting as a spare, it is essential to arrange the refrigerating machinery as shown in Figure VII. In this arrangement, the load on each compressor unit can be maintained independent of the loads on the other compressor units, thereby insuring an equal distribution of lubricating oil to each unit. In other words, the loads for each circuit, that is for circuit No. 1—the M.R. No. 2; Circuit No. 2—the M.R. No. 1 and ice tank, and Circuit No. 3—the B. and E., Vest. and F. and V. compartments can be independently carried by either one of the three (3) compressor units. The controls as indicated in this arrangement consist of a thermal actuated expansion valve and a magnetic solenoid valve in series in the liquid supply line to each circuit.

Description of Equipment:—The refrigerating machinery outfit includes all necessary compressors, condensers, motors, pumps, cooling coils, ice making boxes, control and safety devices, piping including gas dryers, surge drums, etc.

The Navy tonnage rating of the compressor units is based on the following:

A. Compressor suction pressure corresponding to —5 degrees F. evaporator temperature,
B. Inlet water temperature to condenser at 85 degrees F. with not more than 10 gallons per minute of water used per ton of refrigeration.

Compressor:—Compressors are of the vertical, reciprocating, single acting type driven by a direct connected or gear-in-head motor with fly wheel and flexible coupling included in the shafting. Lubrication is of the positive forced feed system supplying oil under pressure to all main, crankshaft, crank pin and wrist pin bearings via drilled crankshafts all supplied by either geared or eccentric type oil pump direct connected or positively driven from crankshaft, and taking suction from adequate sump fitted with oil strainers. The compressor shafts are sealed against gas pressure by the means of a bellows or sylphon seal. The compressor has a minimum gross piston displacement of not less than 20,000 cubic inches per minute per ton. Gas admission is either through the crankcase or through the cylinder, although cylinder or truncated

piston admission is better and more widely used since it provides better oil distribution among multiple units. Compressor speed in general does not exceed 600 revolutions per minute.

Figure VIII illustrates a sectional elevation view of a typical 1-ton Navy type refrigerating compressor.

Condenser and Integral Receiver:—The material of the shell wherein the "Freon-12" is condensed is of either seamless drawn steel or brass tubing, composition "N-r" tube sheets, composition "G" or "M" water heads with 5/8-inch O.D., .049-inch thick copper-nickel tubes, expanded into the tube sheet. The design is based on either U-tubes with separable tube sheet bolted to the shell flange or on straight tubes with brass shell welded to the tube sheets and the condenser heads through bolted to the tube sheets. The amount of cooling surface is based on a water inlet temperature of 85 degrees F. with not over 10 gallons per minute per ton circulated. The condenser is of such size that when the equip-

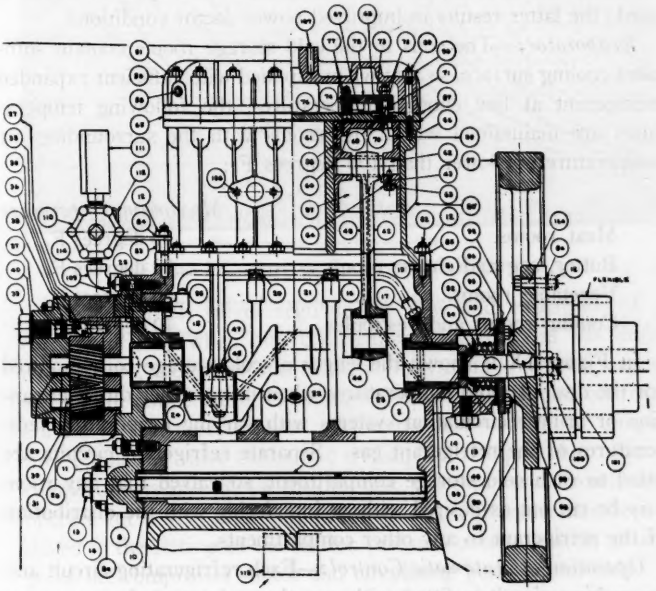


FIGURE VIII.—TYPICAL ONE TON NAVY COMPRESSOR.

ment is operating at specified capacity and under the required conditions, the head pressure on the compressor should not exceed 140 pounds absolute. Furthermore, the size is such that it contains at least 1 square foot of surface for every 1500 cubic inches of piston displacement. The heat transfer rate for condensing "Freon-12" with sea water usually varies between 50-90 B.T.U. per hour per square foot per degree M.T.D. In the event the circulating water is taken from the ship's fire or flushing system, water regulating valves actuated by the "Freon-12" discharge pressure are provided to reduce the quantity of water used to a minimum.

Motor and Control Equipment:—The size of the compressor motor is not less than 3½ horsepower per ton Navy refrigeration. When compressors start without load, normal torque squirrel cage motors are used with full voltage or reduced voltage starting, but for automatic operation of small machines as installed aboard ship, high torque induction motors are used. The motor is either direct connected to the compressor or the gear-in-head type of motor is used; the latter results in improved power factor conditions.

Evaporator:—The coils in the cold storage rooms contain sufficient cooling surface, so that when supplied with sufficient expanded refrigerant at low enough temperatures, the following temperatures are maintained within the rooms with the surrounding air temperature not lower than 100 degrees F.:

	Maximum temperature
Meat rooms	15 degrees F.
Butter, egg and cheese rooms	32 degrees F.
Vegetable rooms	40 degrees F.
Cooling rooms and vestibules	40 degrees F.

In Figure IX is shown the length of cold storage coils required for the desired room temperatures when using either direct expansion or brine recirculation systems with varying evaporating temperatures of the refrigerant gas. Separate refrigerant circuits are fitted to each cold storage compartment, so valved that any room may be cut out as desired without interfering with the distribution of the refrigerant to any other compartments.

Operation of Automatic Control:—Each refrigerating circuit and ice making circuit is fitted with one thermal actuated or pressure

actuated expansion valve in the liquid line and one magnetic solenoid valve in liquid line or temperature regulating valve in the return suction line, both being actuated by a thermostat located in the compartment or submerged in the brine in the case of the ice making box. Hand expansion by-passes are provided in case of failure of the automatic expansion valves. Refer to Figures VI and VII for illustration of the above types of automatic control.

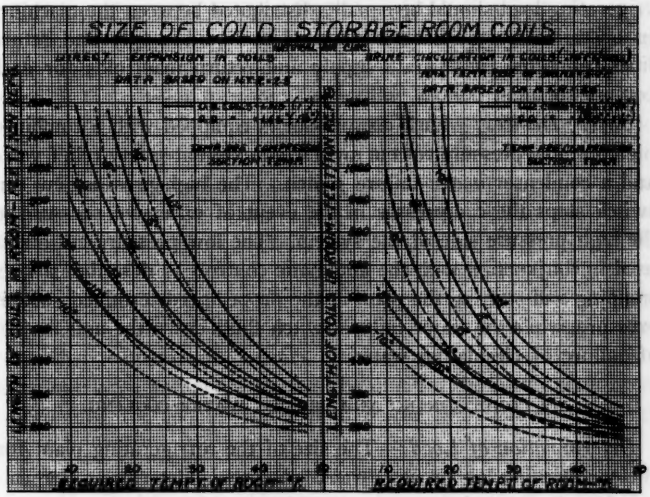


FIGURE IX.—SIZE OF COLD STORAGE ROOM COILS.

The expansion valves regulate the flow and expansion of the refrigerant and when any compartment has reached the desired temperature, the thermostat in the compartment shuts the magnetic solenoid or temperature regulating valve and thereby cuts off refrigeration to that compartment. After the last solenoid or temperature regulating valve closes and the supply of liquid shut off to all compartments and the ice making box, the compressor continues to operate for a very short time until the back pressure has been reduced to the setting of the suction pressure control switch, at which point the compressor motor and circulating pump motor stop. As soon as one of the boxes rises in temperature above the thermostat

setting, the solenoid valve in the supply line or temperature regulating valve in the suction return line of that box will open and allow liquid refrigerant to expand into the cooling coils, which in turn will raise the back pressure above the point of the suction pressure control switch setting, and the compressor unit then starts up again.

When three or more compressor units (Figure VII) are fitted for the same service, with one compressor in reserve, the total refrigerating load should be equally distributed between the machines in operation by the use of independent liquid supply and return lines connected to separate supply and return manifolds for each compressor unit.

Safety Devices:—Each compressor is equipped with a combined high and low pressure cut out switch, an external safety relief valve and a condenser water failure switch. The high pressure cut out switch stops the compressor in the event of excessive pressure in the high pressure side of the system. The low pressure cut out is used in conjunction with the automatic controls as described above, stopping and starting the plant at the proper predetermined suction pressure, while the water failure switch shuts down the unit in case of failure of circulating water.

AIR CONDITIONING AND PROCESS WATER COOLING.

Determination of Load:—The size of the refrigerating plant for air conditioning or process water cooling is determined by calculating the load, in tons, required to perform the required air conditioning or process water cooling.

Plant Arrangement:—Air Conditioning:—A typical arrangement of an air conditioning plant outfitted for operation under reverse cycle conditions, as installed aboard a Navy submarine, is outlined in Figure X. The extra piping, valves and fittings required for reverse cycle operation (air heating during the cold weather periods) and the sequence of valve positions are indicated.

Description of Equipment:—The air conditioning or process water cooling machinery outfit includes about the same type of machinery as required for the refrigerating plant except for minor details.

The Navy tonnage rating of the compressor units for air conditioning or process water cooling is based on the following:

A. Compressor suction pressure corresponding to + 35 degrees F. evaporator temperature,
B. Inlet water temperature to condenser at 85 degrees F., with not more than 10 gallons per minute of water used per ton of refrigeration.

It is of importance to note that a 1-ton unit based on the Navy tonnage rating for refrigeration is equivalent to about 2½ tons based on Navy tonnage rating for air conditioning.

Refrigerating units are provided aboard certain vessels for cooling the process water for the photographic laboratory and X-ray dark rooms. In the former installation, two (2) units are provided in the developing room for cooling motion picture film and cut film tanks. One unit is provided to cool a " still water " tank, lowering the temperature of water in the tank from 90 degrees F. to 65 degrees F. in 8 hours, while the other unit is provided for cooling the " running water " tanks, lowering the temperature of 600 gallons per hour of water from 90 degrees F. to 65 degrees F. In addition, a third unit of about 3 tons capacity is provided in the finishing room for cooling " running water " for the print washers. These units are furnished for the purpose of maintaining better controlled temperatures, besides eliminating the necessity of operating one large unit at all times. The unit for the X-ray dark room is provided for cooling water used in the development of X-ray plates.

Compressor:—The compressors are of similar construction to those described for the cold storage refrigerating plant, except that belt driven, as well as direct connected commercial units are acceptable. The compressor has a minimum gross piston displacement of not less than 8000 cubic inches per minute per ton in lieu of 20,000 as required for the refrigerating plant compressors.

Condenser:—Water cooled condensers are of the same design and construction as outlined for the cold storage refrigerating plant except the cooling surface contains at least 1 square foot for every 600 cubic inches of piston displacement. For small units air cooled condensers of finned tube construction are satisfactory on account of their simplicity, elimination of water connections and due to the fact that it cannot freeze or fail abruptly under reverse cycle conditions.

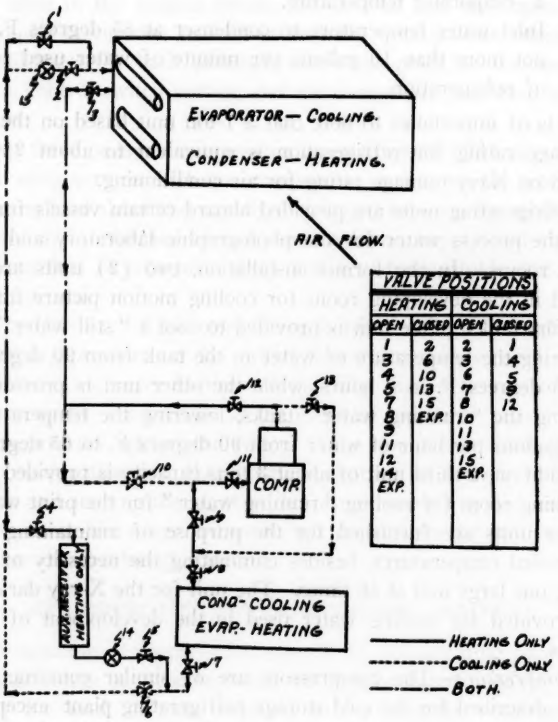


FIGURE X.—COOLING AND HEATING CYCLES.

Miscellaneous Equipment:—The size of the compressor motor should not be less than 1¼ horsepower per ton of air conditioning. Finned tube type evaporators are provided which operate on the basis of forced air circulation across the surface. Automatic control is provided for maintaining specified air conditioning and/or cooling water requirements. The distribution of the refrigerant is by means of expansion valves. In addition to the safety devices specified for the refrigerating plant, an automatic device is required for shutting down the compressor if the system is to be

operated under reverse cycle conditions, when the temperature of the water leaving the condenser is 34 degrees F. or less, in order to prevent a " freeze."

SHIP'S SERVICE STORE EQUIPMENT.

Determination of Load:—In the manufacture of ice cream, refrigeration (direct expansion) is employed in cooling the mix to about 75 degrees F. after pasteurization, in ripening the mix (cooled to about 40 degrees F. and held at this temperature for 24-36 hours), in freezing the mix (enters at about 40 degrees F. and freezing starts at about 29 degrees F.) and in hardening the mix.

Example:—Assume mix enters freezer at 46 degrees F. and is then cooled to 27 degrees F. in the freezer and in the hardening room to 0 degrees F. Then total heat removed from each pound of cream is calculated as follows:

(A) Cooling mix to freezing point:
$1 \times .8\ (46\text{-}27) = 15.2$ where .8 = specific heat of mixture, 27 degrees F. = freezing point.

(B) Latent heat of freezing:
$1 \times 90 = 90.0$ where 90 = heat of fusion.

(C) Hardening of mixture:
$1 \times .45\ (27\text{-}0) = 12.2$ where .45 = specific heat of frozen cream.

Total refrigeration per pound, 117.4.

Refrigeration per gallon $117.4 \times 4.86 = 572$ B.T.U. where 4.86 = weight of 1 gallon of cream based on an over-run of 85 per cent and weight of 1 gallon of mix = 9 pounds.

Plant Arrangement:—The general arrangement of an ice cream and hardening plant for installation aboard an aircraft carrier or capital ship is shown in Figure XI. The capacity of the equipment is indicated on this diagram.

Compressors:—The compressors for the soda fountain, hardening cabinet and ice cream freezer are of the same identical size and designed for a piston displacement of not less than 15,000 cubic inches. It is desirable to crossconnect these units as shown in Figure XI, except that continued operation of both units in parallel

should be avoided and the compressor lubrication system for both units should be carefully watched. The design and construction of the compressors are similar to the compressors for the air conditioning plants except splash lubrication is permitted.

Condenser:—Same requirements as outlined for the condenser for the refrigerating plants, except the surface contains 1 square foot for every 1200 cubic inches piston displacement.

Miscellaneous Equipment:—The motor is of the direct connected or gear-in-head type. An automatic control is provided by a quick-acting pressure switch, operating from the low pressure of the system. Safety devices include the high pressure cut-out and water failure switches. Copper tubing, with compression couplings is employed only in this particular application of refrigeration in view of the limited amount of refrigerant piping involved.

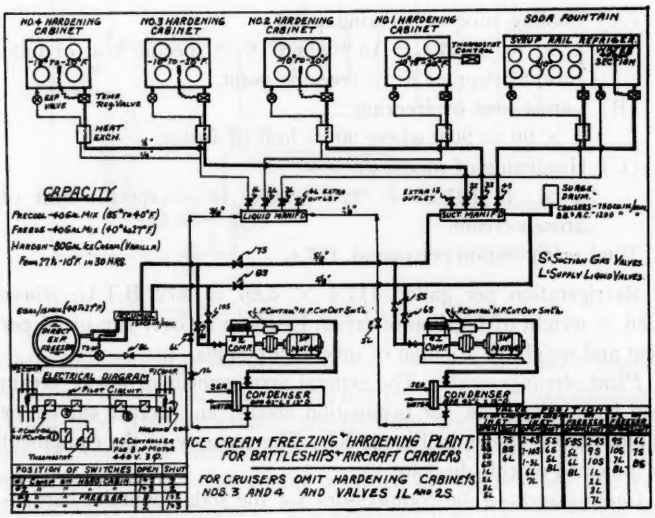
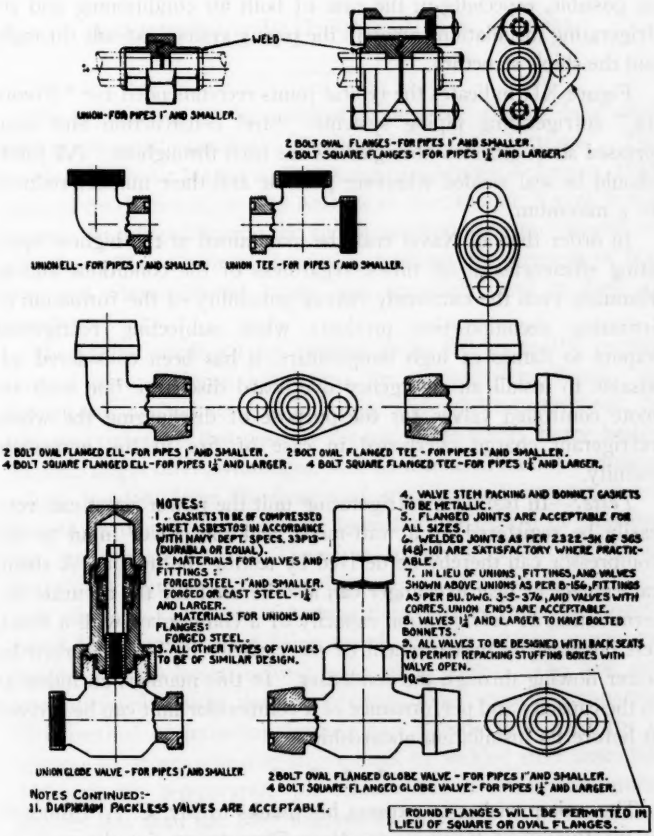


FIGURE XI.—ICE CREAM FREEZING AND HARDENING PLANT.


WELD
UNION - FOR PIPES 1" AND SMALLER.
2 BOLT OVAL FLANGES - FOR PIPES 1" AND SMALLER.
4 BOLT SQUARE FLANGES - FOR PIPES 1¼" AND LARGER.
UNION ELL - FOR PIPES 1" AND SMALLER.
UNION TEE - FOR PIPES 1" AND SMALLER.
2 BOLT OVAL FLANGED ELL - FOR PIPES 1" AND SMALLER.
4 BOLT SQUARE FLANGED ELL - FOR PIPES 1¼" AND LARGER.
2 BOLT OVAL FLANGED TEE - FOR PIPES 1" AND SMALLER.
4 BOLT SQUARE FLANGED TEE - FOR PIPES 1¼" AND LARGER.
NOTES:
1. GASKETS TO BE OF COMPRESSED SHEET ASBESTOS IN ACCORDANCE WITH NAVY DEPT. SPECS. 33P13 - (DURABLA OR EQUAL).
2. MATERIALS FOR VALVES AND FITTINGS:
FORGED STEEL - 1" AND SMALLER.
FORGED OR CAST STEEL - 1¼" AND LARGER.
MATERIALS FOR UNIONS AND FLANGES:
FORGED STEEL.
3. ALL OTHER TYPES OF VALVES TO BE OF SIMILAR DESIGN.
4. VALVE STEM PACKING AND BONNET GASKETS TO BE METALLIC.
5. FLANGED JOINTS ARE ACCEPTABLE FOR ALL SIZES.
6. WELDED JOINTS AS PER 2322-3K OF SGS (48)-101 ARE SATISFACTORY WHERE PRACTICABLE.
7. IN LIEU OF UNIONS, FITTINGS, AND VALVES SHOWN ABOVE UNIONS AS PER B-156, FITTINGS AS PER BU. DWG. 3-S-376, AND VALVES WITH CORRES. UNION ENDS ARE ACCEPTABLE.
8. VALVES 1¼" AND LARGER TO HAVE BOLTED BONNETS.
9. ALL VALVES TO BE DESIGNED WITH BACK SEATS TO PERMIT REPACKING STUFFING BOXES WITH VALVE OPEN.
10.
UNION GLOBE VALVE - FOR PIPES 1" AND SMALLER
2 BOLT OVAL FLANGED GLOBE VALVE - FOR PIPES 1" AND SMALLER.
4 BOLT SQUARE FLANGED GLOBE VALVE - FOR PIPES 1¼" AND LARGER.
NOTES CONTINUED:-
11. DIAPHRAGM PACKLESS VALVES ARE ACCEPTABLE.
ROUND FLANGES WILL BE PERMITTED IN LIEU OF SQUARE OR OVAL FLANGES.

TYPICAL JOINTS FOR "FREON" REFRIGERATING PIPING

FIGURE XII.

MISCELLANEOUS.

Refrigerant Piping Systems:—Extreme care should be exercised in designing a piping system that will eliminate leakage as much as possible, especially in the case of both air conditioning and refrigerating installations wherein the piping system extends throughout the ship's structure.

Figure XII indicates the typical joints recommended for "Freon-12" refrigerating piping systems. Steel construction and compressed sheet asbestos fiber gaskets are used throughout. All joints should be seal welded wherever possible and their number reduced to a minimum.

In order that all Naval craft be maintained at the highest operating efficiencies at all times, regardless of the conditions and to eliminate even the extremely remote possibility of the formation of irritating decomposition products when subjecting refrigerant vapors to flames of high temperature, it has been considered advisable to install an emergency overboard discharge line with remote controlled valves for the purpose of discharging the whole refrigerant charge overboard in case of fire in the immediate vicinity.

Tests:—In testing a refrigerating unit the power input can very easily be registered by a watt-meter and the power input to the compressor can thereby be derived by testing the motor. A steam calorimeter or heat exchanger can also be used for the accurate determination of refrigeration capacity of a compressor with a check test obtained by measurement of the capacity by heat absorbed by water flowing through the condenser. In this manner, an index as to the capacity and performance of a compressor unit can be arrived at before its installation aboardship.

* * * *

The writer wishes to express his thanks to Mr. C. A. Edmonds of the Bureau of Engineering, Navy Department, for the work he has done in the original preparation of some of the graphs contained in this paper.

SILVER SOLDERING BY ELECTRICITY.

By LIEUTENANT P. H. RYAN, U.S.N., MEMBER

The general subject of soldered fittings has already been dealt with in the article entitled, "The Navy's Soldered Fittings." This article, written by Mason S. Noyes, appeared in the February, 1935, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. In this publication the author touched upon the reasons for adopting soldered fittings, the design features, the strength of both soft and silver-soldered joints, and materials used in their assembly.

The means employed for heating these fittings in the production of silver-soldered joints was the oxy-acetylene flame. Although this method resulted in strong, dependable connections, the idea was conceived that possibly the use of electricity as the heating medium might offer advantages, such as increased facility in making joints, freedom from the danger of cracking and warping due to uneven heating, lessened danger from the fire hazard of shipboard use, and perhaps others. Therefore, an investigation was authorized by the Bureau of Engineering.

A number of methods of heating these fittings have been investigated, the materials used being mainly copper tubing, composition flanges and the lower melting point silver solder. The first steps were directed toward the development of an induction apparatus. This method of induction heating had already been used for the purpose of stress-relieving larger sizes of welded steel assemblies. Although the temperature for stress-relief is some two hundred degrees less than the 1300 degrees F. necessary to melt efficiently the low silver-content solder, and the heating of copper and bronzes to higher temperatures is more difficult than the heating of steel, due to the greater heat conduction of the former, it was decided to attempt this method first.

The original apparatus consisted of a closely-coiled helix, formed

of flattened copper tubing, connected to the terminals of two parallel 30 kva 220-30 volt transformers. This coil was water cooled. It had a total of eight turns. This apparatus showed promise, in that an assembly consisting of a 6-inch extra heavy copper tube and a composition flange were brought to a sufficiently high temperature to melt the solder at the joint. With this promising result at hand, it was then decided to construct a hinged coil, since in the fabrication of assemblies it would be necessary to have a more or less flexible apparatus that could be placed closely about the hub of a flange and subsequently removed. Therefore, another coil was constructed of flat strip brass, so hinged that it could easily be placed around and removed from a flanged assembly. Since this could not be water cooled, provision was made to cool it with a draught of air. Difficulties arose due to the fact that the power necessary and the time required were excessive. Its advantage over the oxy-acetylene method in that regard was, to say the least, questionable. Therefore, it was decided to continue the investigation of other methods.

Next on the list was a design employing resistance rather than induction. It might be termed an indirect resistance heating method since the resistance unit was imbedded in refractory material and the heat brought to the flanged assembly by conduction. The resistance element was made in two half-circles, fashioned in the form of a Roman frieze and hinged together to permit placing about the work assembly. This method lacked dependability, due to the difficulty of obtaining a material of sufficient strength at high temperatures to withstand the unavoidable stresses imposed upon it. Also, the time element entered the problem in such degree as to render it of doubtful advantage.

The third method, and it may be said the apparently successful one for silver soldering the larger-sized flanged assemblies, will be described in more detail than those previously mentioned. It is a direct resistance heater wherein the flange and tube assembly constitutes a part of the electrical circuit. This apparatus has been brought to its present state of development by one department of a large commercial concern.* The apparatus is a modification of a standard rivet heater.

* The Berwick Heater Dept. of the American Car & Foundry Co.

Figure 1 is a sketch giving the general design features of the direct resistance heater. As may be discerned from the plate, the apparatus is primarily a single-phase transformer, the secondary of which consists of a single turn only, made up of copper sheet laminations. The secondary terminates in heavy copper blocks,

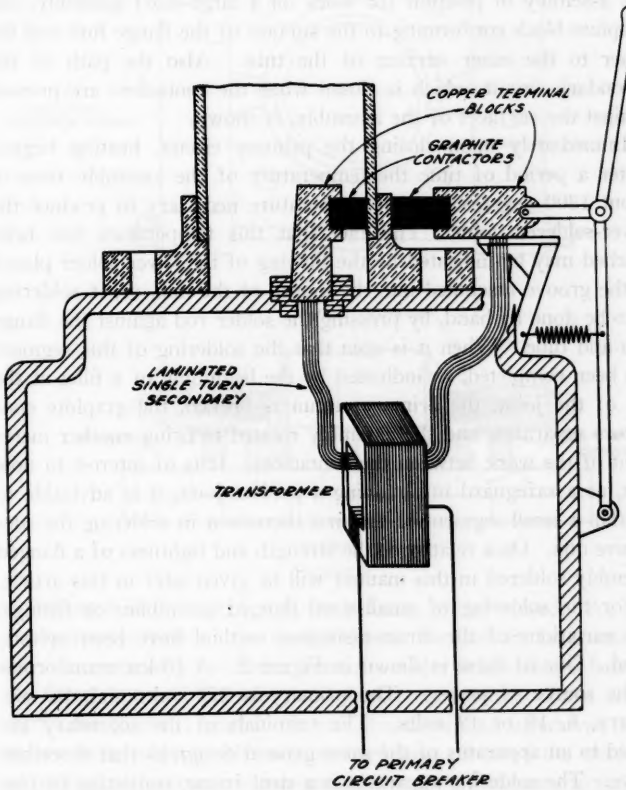


FIGURE 1.

which are movable in relation to each other. The relative position of these blocks is controlled by means of a system of levers pivoted on the frame of the transformer housing. Thus it is seen that this movement facilitates work on assemblies of any size up to the

maximum that can be introduced with the blocks in their most widely separated position. This system of levers permits holding the blocks in any desired position. The copper blocks are equipped with graphite contactors. These contactors are shaped to conform to the contact surfaces of the work to be soldered. Plate 1 shows the assembly in position for work on a large-sized assembly, one graphite block conforming to the surface of the flange hub and the other to the inner surface of the tube. Also the path of the secondary circuit, which is closed when the contactors are pressed against the surfaces of the assembly, is shown.

Immediately upon closing the primary circuit, heating begins. After a period of time the temperature of the assembly rises to about 1300 degrees F., that temperature necessary to produce the silver-soldered joint. The fact that this temperature has been reached may be indicated by the flowing of the silver solder placed in the groove machined into the mouth of the hub, or, if soldering is to be done by hand, by pressing the solder rod against the flange hub and tube. When it is seen that the soldering of this segment has been completed, as indicated by the beginning of a fillet at the top of the joint, the primary circuit is broken, the graphite contactors separated, and the assembly rotated to bring another increment of the work between the contactors. It is of interest to note that, as a safeguard in obtaining a perfect joint, it is advisable to overlap a small segment of the first increment in soldering the successive one. Data relative to the strength and tightness of a flanged assembly soldered in this manner will be given later in this article.

For the soldering of small-sized flanged assemblies or fittings, two variations of the direct-resistance method have been investigated. One of these is shown in Figure 2. A 10-kw transformer is the source of power. The primary is 220 volts and the secondary, 8, 10 or 12 volts. The terminals of the secondary are bolted to an apparatus of the same general design as that described above. The soldering apparatus is a steel frame, consisting of two bridge members and two tie rods well insulated from each other. The bridge members support the graphite contactors, as indicated in the sketch on Figure 2. The contact faces of the graphite blocks are one inch high and ½ inch wide, the ½-inch dimension being the peripheral one. The soldering device is placed about the flange,

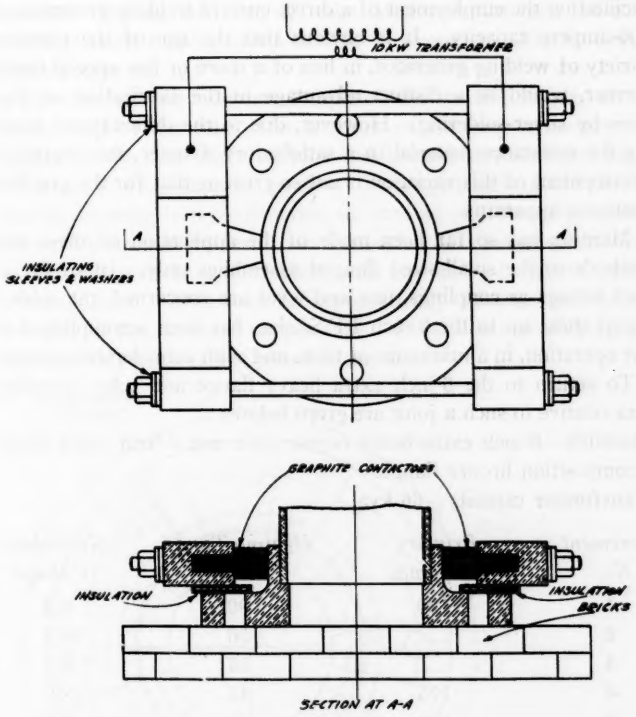


FIGURE 2.

as indicated in the sketch, and the primary circuit closed. As soon as the temperature of the work between the contact faces reaches the value necessary to effect a joint, the solder is introduced between the flange hub and the tube; and the circuit broken. Then an adjacent increment is heated. It was found that, as successive increments were soldered, the accumulated heat obviated the necessity of making each successive increment immediately adjacent to the preceding one, and that, before the completion of the entire periphery, the spacing between increments could be gradually increased. Further particulars will be given concerning a joint made by this method.

The variation of this method differs only in the particular that a higher resistance material of a proprietary nature is used,

facilitating the employment of a direct-current welding generator of 400-ampere capacity. It is obvious that the use of the common variety of welding generator, in lieu of a more or less special transformer, would be a distinct advantage in the fabrication of pipe lines by silver-soldering. However, due to the difficulty of forming the resistance material in a satisfactory manner, the degree of development of this variation is not so great as that for the graphite contactor apparatus.

Mention has so far been made of the application of these two methods to the small-sized flanged assemblies only. In so far as such fittings as couplings, tees and wyes are concerned, the soldering of these up to the 2-inch I.P.S. size, has been accomplished in one operation, in a minimum of time, and with considerable success.

To return to the 6-inch extra heavy flange and tube assembly, data relative to such a joint are given below:

Assembly—6-inch extra-heavy copper tube and 6-inch extra-heavy composition bronze flange.

Transformer capacity—66 kva.

Increment No.	*Primary Amperage*	*Heating Time, Secs.*	*Secondary Voltage*
1	120	90	8.2
2	.....	50	8.2
3	.....	35	8.2
4	175	35	8.2
5	110	30	6.8
6	110	30	6.8
7	110	30	6.8
8	100	30	6.2
9	100	30	6.2
		Total, 360	

This tube was received "as soldered." In order to facilitate testing it hydrostatically and in tension, a steel disc was brazed into the end of the tube.

The assembly withstood 600 pounds per square inch of pressure (giving a hoop stress of 6000 pounds per square inch) for seven

minutes. During this time no leaks developed, although the joint was vigorously tapped with a machinist's hammer. The fluid used was kerosene oil.

The tension test was not completed due to the failure of the brazed end of the tube. As a result of the heat necessary for brazing in the steel disc, the tube end became annealed and flared out when it was loaded in tension. This apparently subjected the brazing material to a combination of bending and shearing stresses beyond its strength to resist, since it broke. However, a total load of 168,400 pounds was reached with no sign of failure in the soldered joint. This load is approximately 66 per cent of the strength of the annealed copper tube. Plate I is a photograph of this assembly after it was tested in tension.

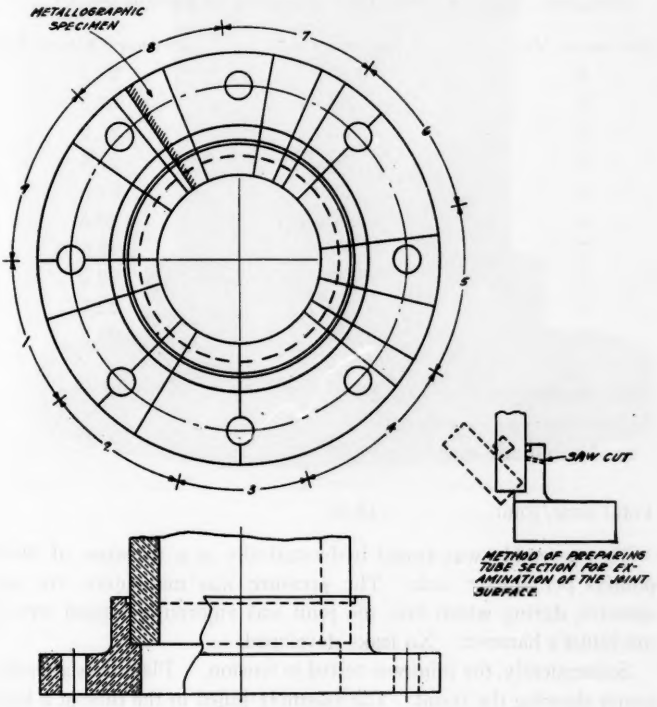


FIGURE 3.

Subsequent to this tension test, the soldered assembly was sectioned, as shown in Figure 3, by cutting radial slices from the tube and flange. These cuts were made in such locations as to show the extent of the bond both within the arc of an increment and in the inter-increment zones. It was found that, with the exception of one cut which obviously contained foreign matter throughout ⅔ of its depth, a remarkably large percentage of the joint was bonded. The bond was so strong that in a few cases the flange section was ruptured in the attempt to strip off the tube section.

Data concerning a test of the device for soldering the small-sized flanged joints and fittings are given below:

Assembly—2 4-inch I.P.S. standard flanges of composition "G."
1 4-inch I.P.S. copper tube, 14 inches long × .109-inch wall thickness. Silver solder (1300 degrees F. soldering temp.).

Increment No.	*Time, minutes*	*Primary power, kva*
1	6.40	22.5
2	.65	22.5
3	.35	22.5
4	.50	22.5
5	.60	22.5
6	.90	22.5
7	.70	22.5
8	.75	22.5
9	.70	22.5
10	.65	22.5
Total heating time	12.20	
9 inter-increment periods at .15 minutes each	1.35	
Total time/joint	13.55	

This assembly was tested hydrostatically at a pressure of 2000 pounds per square inch. The pressure was maintained for ten minutes, during which time the joint was vigorously tapped with a machinist's hammer. No leaks developed.

Subsequently, the joint was tested in tension. Plate II is a photograph showing the result. The assembly failed in the tube at a load

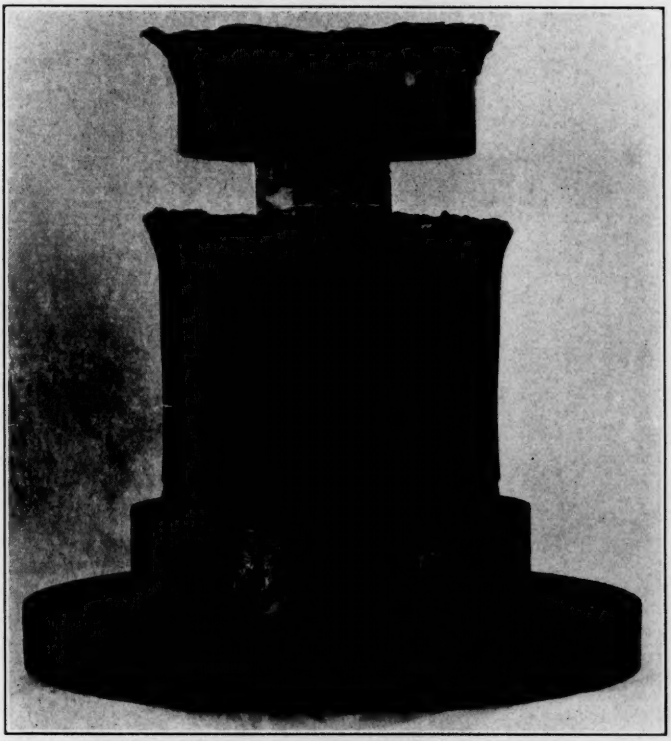


PLATE I.

PLATE II.

of 47,825 pounds, which is a unit stress of 31,700 pounds per square inch for the tube. Later the joint was sectioned as indicated for the 6-inch extra-heavy tube. However, since the tube sections failed in bending when it was attempted to remove them from the flange sections, no examination could be made of the bond other than along the edges of the sections. This examination indicated that the silver solder had penetrated to the root of the joint.

A word should be said here relative to the effect of electrical silver-soldering on the microstructure of the materials soldered. There was no evidence of overheating in the 6-inch extra-heavy flange, as the grain size was very uniform throughout, that in the hub being no larger than that in the face section. Also, there was no apparent difference between the grain size in the heated zone of the tube and that in the part of the tube at a distance from the heat zone. In the case of the 4-inch flange, no marked difference was noted. However, the heat zone of the tube did reveal grain growth, though not of large degree.

As was stated earlier, this method may be classified as direct resistance heating, as differentiated from the method wherein the resistance element is not in direct contact with the materials to be soldered. However, attention should be drawn to the fact that these heaters differ from resistance welders and heaters used for other purposes in that the heat in the subject assemblies is developed mainly in the contact pieces. In resistance welders the effective resistance is the contact resistance of the parts to be united. In heaters such as rivet heaters, the object to be heated is usually the effective resistance. In this method, herein described, the larger part of the effective resistance is in the contact pieces, which heat the work by conduction. The graphite contactors are so formed that their cross-section offers its greatest resistance adjacent to the contact areas. Thus it is possible to handle relatively large volumes of metal with a comparatively small capacity apparatus.

The results obtained with the electrical devices for silver-soldering indicate that the process is practicable. It appears that the above-described apparatus may be developed to such a degree as to render their use on a large scale in the field feasible and advantageous. However, in order to warrant the adoption of a new method or procedure it must be shown that, in the final analysis, it offers advantages over current methods or procedures.

To begin with, the strength and tightness of the joints obtained are comparable with those obtained by the use of the oxy-acetylene flame. There appears to be no great choice in the matter from that standpoint since both result in joints of greater strength than that of the standard sized tube material.

Experience with the electric method points to the possibility of regularly obtaining pressure-tight, full-strength joints, in one cycle of operation. The ease with which this has been accomplished offers a distinct advantage over results obtained with the use of oxy-acetylene gas, since patch-work has been found to be necessary at times with the latter method.

Examination of the microstructure has revealed that this method has no apparent effect upon the grain structure of either the heavy copper tubing or the flange material. The effect upon the thinner tubing, the particular set-up for which required the longer heating period, was not serious as it resulted in only a partial anneal. The shorter the period of time required for the operation, the smaller is the heat effect upon grain structure.

Another advantage lies in the fact that the danger of overheating, with its undesirable results, is reduced. For the smaller sized joints, which may be made in a single operation, the problem of unequal heat distribution is obviated. This results in less tendency toward warping and cracking.

Lastly, based upon the time required to produce joints by the two methods and upon the cost of power and that of fuel, the advantage lies with the electrical method from the standpoint of economy.

The use of an apparatus which can be controlled with a minimum of effort on the part of the operator, tends to decrease the effect of the human element error. The operation of these devices is not difficult and should require neither exceptional intelligence nor undue exertion.

Whereas the soldering of large sized fittings is, and perhaps will remain, a shop job, it is not at all inconceivable that fabrication of small and medium sized lines may be done either at the ship's side or "in place" on board the ship itself.

BOW CHARACTERISTICS FOR ICE BREAKING.

By LIEUTENANT D. R. SIMONSON,* U.S.C.G.

The increased use of oil as fuel for private heating plants has resulted in a rapid growth of medium size marine carriers especially of tanker and barge design. Since the peak demand for this type of cargo comes in the severest part of the winter, and since a large portion of their voyages may be in fresh waterways often found closed by ice, navigation depends on the ability of the vessel to break ice or on the assistance of an ice breaker.

There are several factors determining the ice breaking qualities of a vessel which must be balanced against cargo capacity, steering and maneuvering qualities, drafts allowable, and economical power. The main factors determining the ice breaking characteristics of a vessel are (1) water line length, (2) normal displacement of vessel, (3) thrust available from the propeller at low speeds, and (4) fore body design.

The length and displacement of a vessel are usually fixed by the prime service for which the craft is intended. Thrust will be limited by propeller and power plant efficiency as well as the total horsepower considered desirable to install. The efficiencies may lie between wide limits for the various power plants, propeller sizes and spreads available, and are somewhat difficult to determine without comparison with similar installations or without results of model basin tests; however, approximations are included herein which should give reasonably dependable figures.

The tensile strength of ice varies with the impurities contained therein, nature of formation, age and temperature. The laboratory experiments of the University of Illinois in 1895 (The Technograph, Vol. IX) gave the tensile strength of ice as 102 to 256 pounds per square inch for temperatures of 19 degrees to 23 degrees Fahrenheit. The lower figure is taken as strength of salt water ice and the higher one as the strength of fresh water ice.

* Constructor, U. S. Coast Guard.

The formula for resistance of ice to stress by a concentrated load has been developed from one (see formula No. 1) presented by Slocum & Hancock for flat plates where a homogeneous circular form of thickness (t) supports a concentrated load (w), which is distributed over a small radius (r_o) concentric with the plate; the circular plate freely resting on a circular rim of radius r.

$$(1)\ \text{Stress } S = \frac{3W}{\pi t^2}\left\{1 - \frac{2 r_o}{3 r}\right\}$$

$$\text{Where } r_o \text{ is small compared to } r,\ \frac{(2 r_o)}{(3 r)} = o$$

$$(2)\ \text{then } S = \frac{3 W}{\pi t^2}$$

therefore the thickness of ice of tensile strength S that can be broken by force W, neglecting reserve buoyancy of ice, is

$$(3)\ t = \sqrt{\frac{3 W}{\pi S}}$$

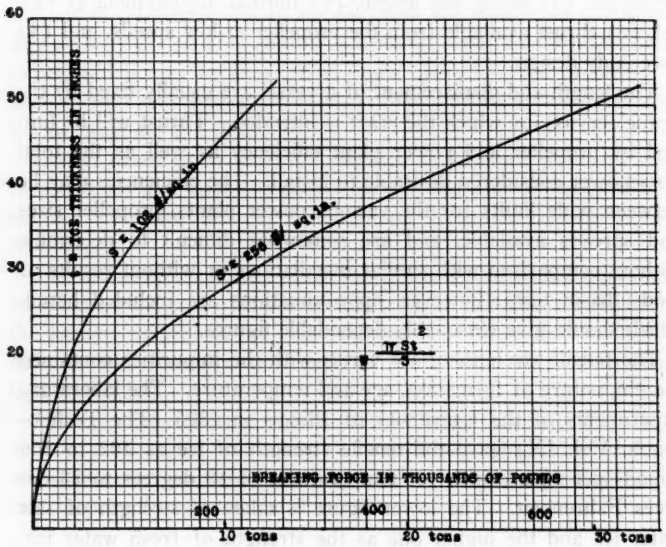


FIGURE 1.

The results of this equation have been plotted as Figure 1, using 102 and 256 for the breaking stress S.

The force available from a given vessel for breaking ice consists of the vertical reaction W due to the trimming moment, M, of the vessel when forced out of her normal waterplane by the thrust, T, of the propeller. This can be expressed by the approximate formula

$$(4)\quad W = \frac{2240\ M\ Y}{D}$$

where Y = allowable trim in inches and D = distance from center of flotation to portion of stem in contact with ice. This is approximately one-half the waterline distance, L, when the bow is not cut away too much and the change in trim, $\emptyset$, is small (usually should be below 5 degrees). The moment to change trim one inch can be expressed as

$$(5)\quad M = \frac{\Delta\ G\ M}{12\ L}$$

where Δ = displacement in tons, and GM = longitudinal metacentric height. This is approximately equal to the waterline length, L, for normal designs in which case

$$(6)\quad M = \frac{\Delta}{12}$$

then substituting in formula (4),

$$(7)\quad W = \frac{2240\ \Delta\ 12\ L\ \tan\ \emptyset}{12\ L/2} = 4480\ \Delta\ \tan\ \emptyset$$

The thrust available to force the vessel out of trim $\emptyset$ degrees can be approximated by any of several formulae, assuming the line of thrust is coincident and parallel to the top surface of the ice field. The momentum of the ship is neglected as it is desirable to break the ice without charging or ramming.

$$T = \frac{HP \times f \times 33000\ \text{ft. lbs./min.} \times 60\ \text{min./hrs.}}{K \times 6080\ \text{ft/min.}}$$

$$(8)\quad T = \frac{325.7 \times HP \times f}{K}\ \text{pounds,}$$

where HP = total horsepower available less amount necessary to drive ship at speed K.

f = overall efficiency of power plant and propeller at speed K when developing maximum horsepower; varies between 10 and 25 per cent.

K = Velocity expected through ice; about 3 knots.

Another approximation is one ton thrust per 100 I.H.P. at low speeds; then

(9) $T = \text{I.H.P.} \times 22.40$ pounds, or from model basin tests,

(10) $T = C_t N^2 P^2 d^2$

where C = thrust coefficient

N = revolutions per second obtainable at speed K at rated horsepower.

P = propeller pitch in feet.

d = propeller diameter in feet.

The limiting angle, Y, between stem and water plane, permitting the thrust, T, to hold the vessel out of trim $\emptyset$ degrees against the vertical force, W, as represented by Figure (2), is solved graphically in Figure (3). These limiting angles vary as the ship is forced out of trim and may be expressed as tangents to the stem profile as follows:

(11) $Y = (\Theta - \emptyset)$ where Θ equals the angle between the stem and surface of ice field for trim $\emptyset$ and

(12) $\tan \theta = \frac{T}{W}$ and by substituting equations 7, 8, and 12 in 11 we have

$$(13)\quad Y = \tan -1 \left\{ \frac{.0727 \times HP \times f \cot \emptyset}{\Delta K} \right\} - \emptyset$$

from which the bow profile can be plotted tangent to these slopes or by substituting in equation (11) and integrating, the formula

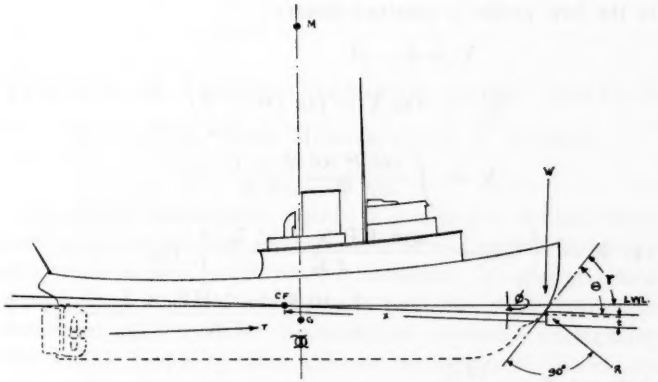


FIGURE 2.

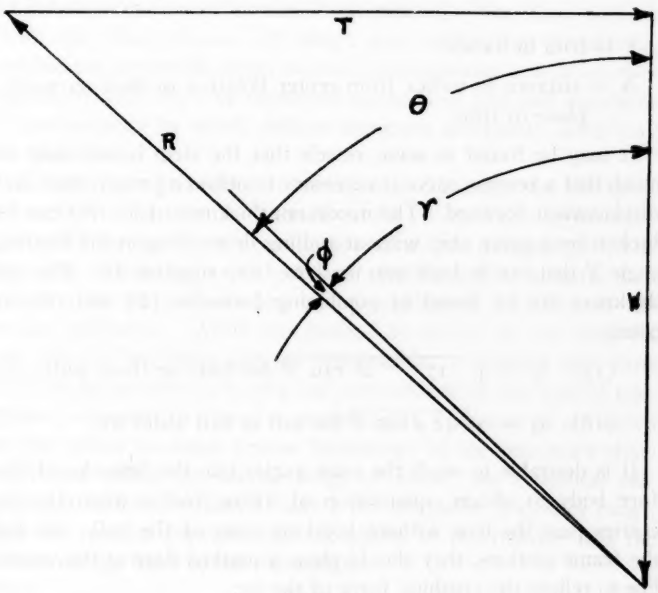


FIGURE 3.

for the bow profile is obtained direct:

$$Y = \Theta - \phi$$

$$\frac{dy}{dx} = \tan Y = \tan (\theta - \phi)$$

$$X = \int \frac{\cot \theta \cot \phi + 1}{\cot \phi - \cot \theta} \, dy$$

$$(14)\quad X = \left\{ 6\,L + \frac{.436\ HP \times f \times L}{\Delta\ K} \right\} \log \left\{ \frac{10.5\ L^2 \times HP \times f}{\Delta\ K} - Y^2 \right\}$$

where $\cot \phi = \frac{12\ L}{Y}$

$$\cot \theta = \frac{W}{T}$$

Y = trim in inches

X = distance in inches from center flotation to stem on water-plane of trim.

It may be found in some vessels that the stem is cut away so much that a reverse curve is necessary to obtain a proper draft and displacement forward. The maximum thickness of ice that can be broken by a given ship without stalling depends upon the limiting angle Y that can be built into the bow from equation 13. This ice thickness can be found by combining formulae (7) and (3) to obtain

$$(15)\quad t_1 = \sqrt{17\ \Delta \tan \phi}$$ for hard or fresh water ice

$$(16)\quad t_2 = \sqrt{42\ \Delta \tan \phi}$$ for soft or salt water ice.

It is desirable to work the same angles into the buttocks of the fore body to obtain equalization of lifting forces when the ice carries pass the bow without breaking clear of the hull. As for the frame sections, they should show a marked flare at the water-line to relieve the crushing force of the ice.

TESTING OF TELEPHONES FOR NAVAL SERVICE.

By C. Huey,* Civil Member, and C. E. Fraser.†

Any speech transmission system, if it is to give faithful reproduction, should transmit and reproduce all the audible frequencies of a sound in their proper relative intensities. To give acceptable reproduction, it should transmit and reproduce those frequencies considered most necessary for minimum acceptable intelligibility. The audible frequency range depends upon physical factors—the frequency-amplitude characteristics of a sound, the characteristics of the masking sound, and the hearing characteristics of the average ear—whereas the acceptable frequency range must be determined by judgment in view of the conditions imposed by the Naval service. With the disappearance of design and engineering limitations, audible and acceptable limits increase in importance and it becomes imperative that there be developed methods of test and standards of performance by which definite minimum acceptable limits may be established.

Formerly the procedure in testing was to average the ability of a number of consecutive listeners at the receiver end to correctly transcribe the words or phrases spoken into the transmitter. The test was thus a matter of personal opinion which, in turn, is a function of the physiological and psychological condition of the testing personnel. While this method of testing is very common and affords important general information, it throws very little light on the problem as to why the performance of one type of telephone is superior to that of a second type. This may be attributed to the failure to obtain precise knowledge of the composite characteristics within the audible range of sound, the acoustical efficiency at various frequencies within the acceptable frequency range, and the relation between audio sensation at the receiver and the sound energy input to the transmitter.

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To afford a method of measurement and analysis of speech transmission through an electro-acoustical system without depending upon the personal element would necessitate some means of readily simulating the voice and ear, and from which may be obtained input and response measurements based on some definite numerical units. To establish a method of test based on absolute units which would accurately depict energy variations from 0.005 microwatt to 7500 microwatts at audio frequencies would necessitate elaborate and complicated test equipment and procedure which was deemed as unwarranted for the results to be obtained. For this reason the method, as finally developed, is a comparative test based fundamentally on the average response of the human ear. Measurements of the relation of reproduced frequency range to intelligibility, as judged by a number of experienced listeners are used in establishing the acceptable frequency range of 500 to 4000 cycles per second.

To eliminate the human voice as the speech generating agency there was developed an electric oscillator (see Plate I), which is in effect a variable frequency alternator. Thus electrical power excites what is essentially a loudspeaker unit of the moving coil type (Plate II). In order to eliminate the characteristic response of the loudspeaker and to obtain a source of sound which will convert the electrical energy input to sound energy output without distortion of wave form or amplitude, a special acoustic loading system is placed over the orifice of the loudspeaker. This permits a wave-form and amplitude of acoustic output which, for any fixed value of input voltage, is independent of frequency. In this way a sound generator is established, of flat characteristic, that can be used as an acoustic standard for the excitation of a telephone transmitter. This unit is arbitrarily termed the artificial voice, not because its acoustical characteristics are similar to the human voice, but because its output at any frequency can be controlled. It is more properly an acoustic generator of stable and reproducible characteristics.

The response indicator, which simulates the human ear, consists of an electro-acoustical system which duplicates the acoustical characteristics of the human ear, and indicates the response in terms of electrical units. It consists of a cavity molded in soft rubber and which is anatomically the average by measurement of

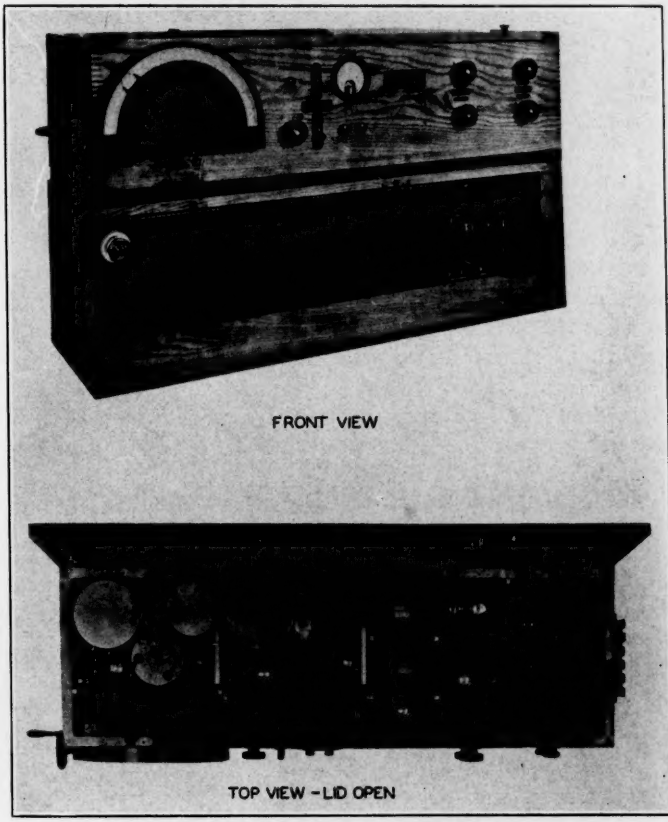

PLATE I.—ELECTRIC OSCILLATOR SINE-WAVE ALTERNATOR.

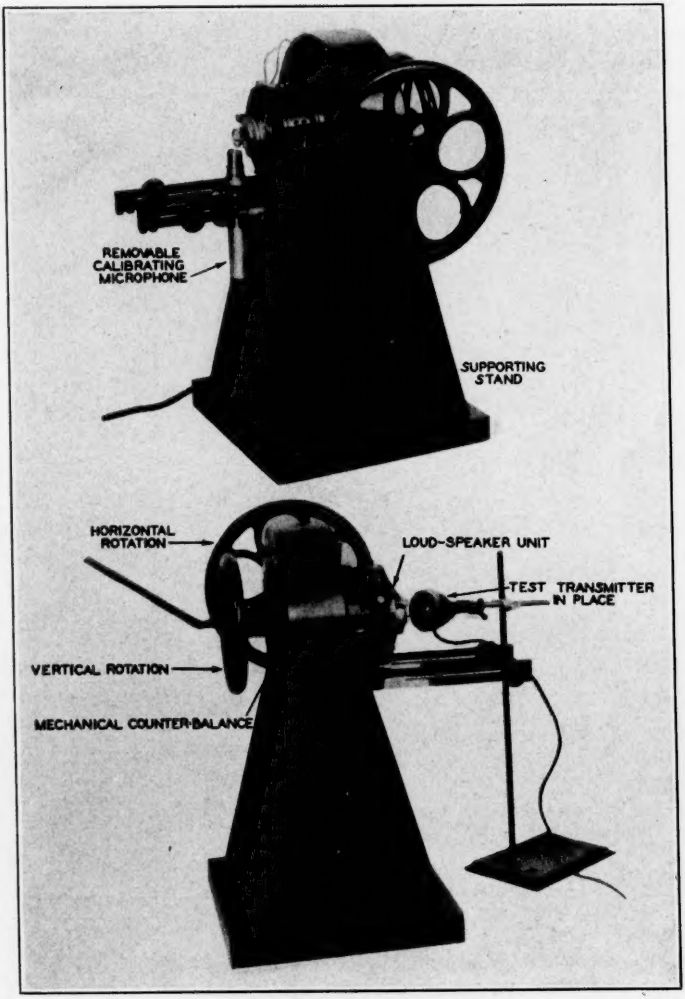

PLATE II.—ARTIFICIAL VOICE ASSEMBLY.

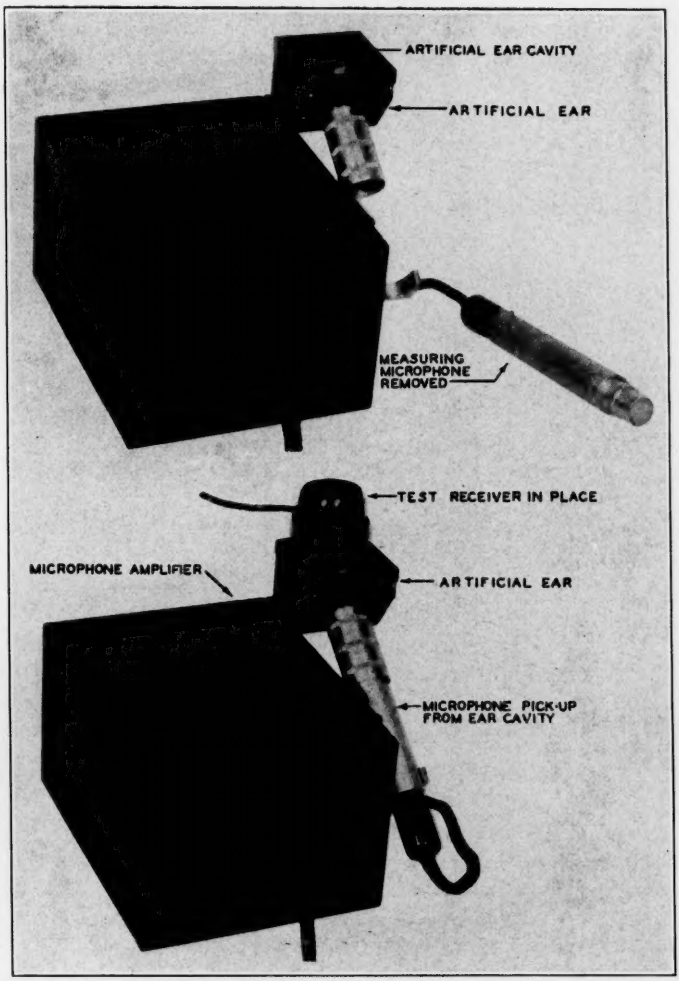

PLATE III.—ARTIFICIAL EAR ASSEMBLY.

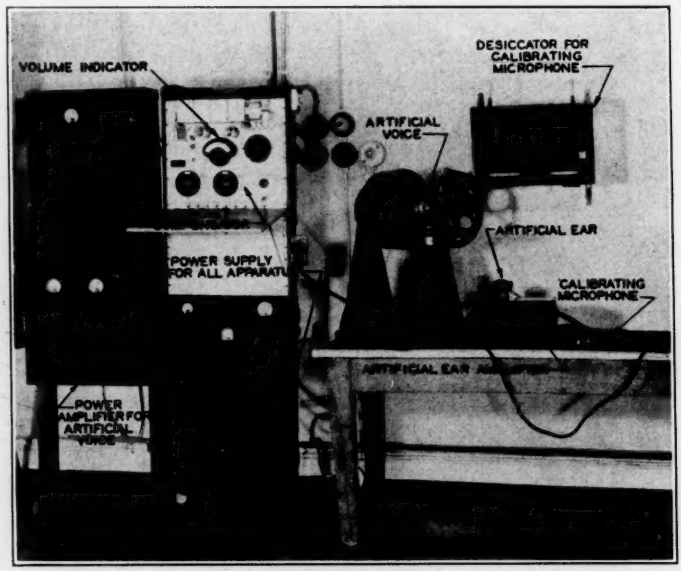

PLATE IV.—ASSEMBLY OF TELEPHONE TESTING DEVICES EXCLUDING OSCILLATOR AND WARBLE FREQUENCY UNITS.

many human ear cavities. It is acoustically filtered, vented and weighted to modify the output of a telephone receiver unit placed thereon, so that the sound in this cavity is modified as to amplitude with reference to a certain level at 1000 cycles per second to simulate the characteristics of this sound as existing in the human ear canal. A small condenser microphone, replacing the tympanic membrane, is activated by the sound pressures within the cavity, and through suitable amplification indicates these values by means of electrical meters (see Plates III, IV and IVa).

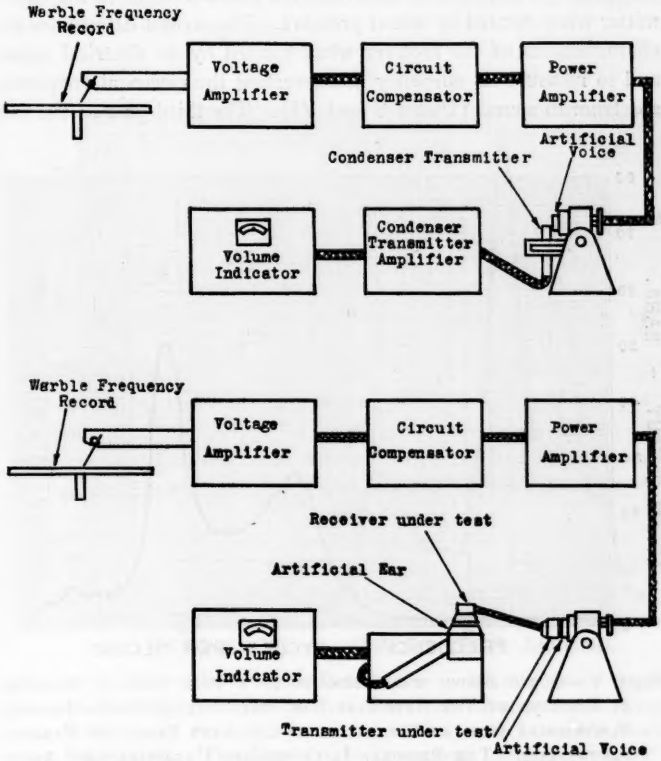

PLATE IVa.—SCHEMATIC DIAGRAMS FOR THE CALIBRATION OF THE ARTIFICIAL VOICE AND EAR.

Having thus eliminated the personal element from the test, there is established a method whereby the performance of the telephones may be indicated as definite numerical values. In order to adequately determine the relative acoustical, electrical, and mechanical performance of the telephone as a unit, and also that of the component parts, the telephone is subjected to the following seven separate tests:

1. *Frequency characteristic test.* The frequency characteristic test of telephone units comprises three separate and distinct determinations. The first is to determine the characteristics of the transmitter when excited by sound pressure. The second determines the characteristics of the receiver when excited by an electrical input, and to measure its efficiency in converting the electrical frequency spectrum to sound (Plates V and VI). The third part of the test

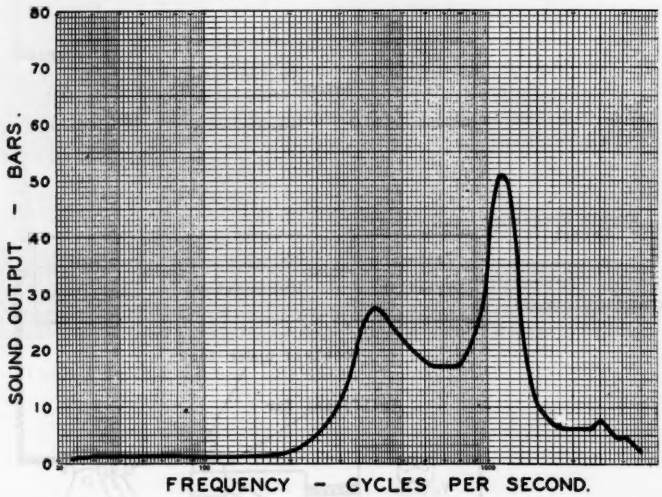

PLATE V.—CURVE SHOWS THE RESPONSE OF A POOR FIDELITY RECEIVER AS RECORDED BY THE ARTIFICIAL EAR, THE RECEIVER BEING EXCITED ELECTRICALLY FROM AN OSCILLATOR AT CONSTANT LEVEL FOR VARIOUS FREQUENCIES. THE RESPONSE IS CONSIDERED UNSATISFACTORY ABOVE 1400 CYCLES.

determines the over-all characteristics, wherein a transmitter is excited by sound energy and the sound output of the telephone receiver is measured (see Plate VII). The artificial voice, or speech generator, excited to produce a constant sound-level output, is used in all cases to drive the transmitter. The response indicator is used to measure the output of the receiver. Thus standard conditions of sound input and sound output measurements are used and the resultant data compared in all cases against the input, whether electrical or acoustical. The conversion characteristics of the telephone equipment are thus obtained and are available for comparison tests of different telephones.

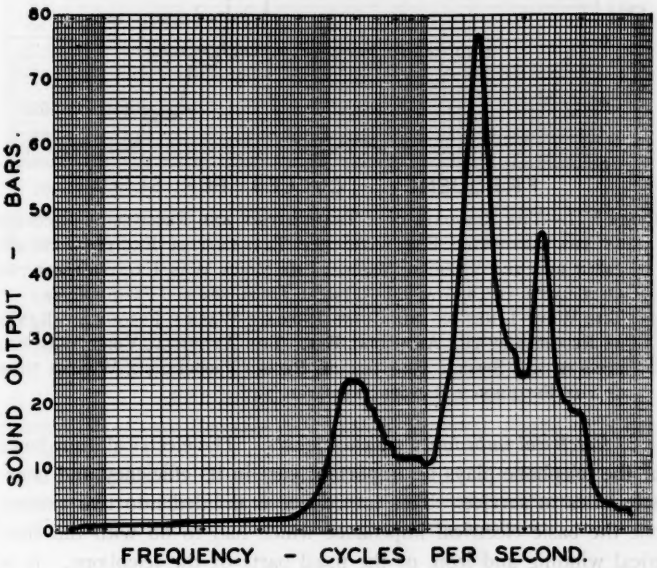

PLATE VI.—CURVE SHOWS THE RESPONSE OF A HIGH FIDELITY RECEIVER AS RECORDED BY THE ARTIFICIAL EAR, THE RECEIVER BEING EXCITED ELECTRICALLY FROM AN OSCILLATOR AT CONSTANT LEVEL FOR VARIOUS FREQUENCIES. THE RESPONSE IS CONSIDERED SATISFACTORY FROM 500 TO 3200 CYCLES.

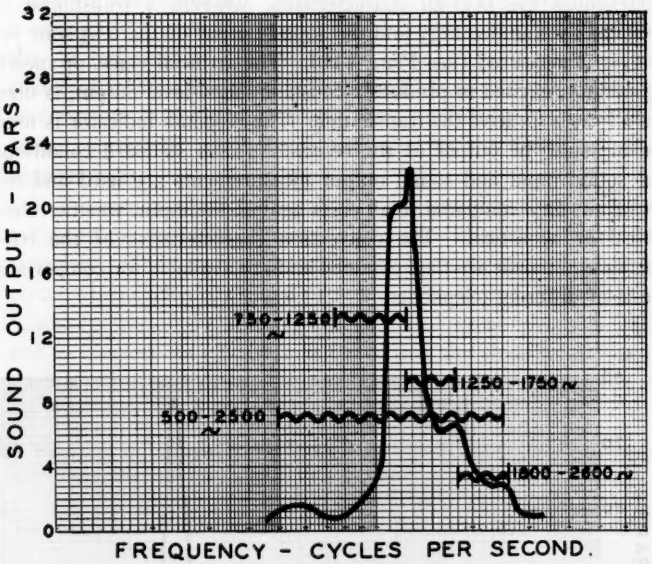

PLATE VII.—THE RESPONSE OF A TRANSMITTER AND RECEIVER AS RECORDED BY THE ARTIFICIAL EAR WITH THE TRANSMITTER EXCITED ACOUSTICALLY BY THE ARTIFICIAL VOICE. THE VOICE WAS EXCITED FROM THE OSCILLATOR AT CONSTANT ELECTRICAL LEVEL FOR ALL FREQUENCIES. THE WAVY LINES ON THE CURVE INDICATE WARBLE FREQUENCY BANDS AS RECORDED BY THE ARTIFICIAL EAR. THE ARTIFICIAL VOICE WAS EXCITED FROM A SPECIAL PHONOGRAPH RECORD PRODUCING A WARBLE NOTE OF THE FREQUENCY RANGES INDICATED.

2. *Electrical impedance measurements.* The electrical impedance of the telephone plays an important part in its efficiency and characteristics in service. This impedance is divided into two classes, one the basic electrical impedance which had to do with the electrical winding and iron, or the fixed parts of the telephone. It is measured with all moving parts blocked so that the magnetic reactions and leakage path of the air gap flux remains constant. This requirement is specified although it has little bearing on the acoustic performance of the telephone. Having once determined, however, that a telephone is fundamentally satisfactory, and other

features of construction and design being standardized, this measurement will give an indication of the uniformity of material and construction between units. The impedance is also measured at many points at different exciting frequencies and a frequency impedance curve is plotted (see Plate VIII). The frequency range used in all testing is normally 500 cycles per second to 4000 cycles per second, this being the acceptable frequency range of an instrument properly to transmit intelligible speech with reasonable fidelity.

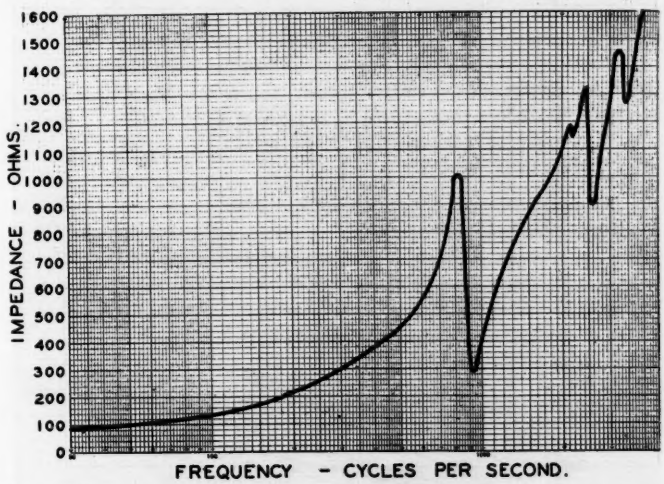

PLATE VIII.—CURVE REPRESENTS A TYPICAL MOTIONAL IMPEDANCE CHARACTERISTIC, BEING THE ELECTRICAL MEASUREMENT OF IMPEDANCE AT EACH FREQUENCY OVER THE VOICE RANGE, WITH A RECEIVER MOUNTED ON THE ARTIFICIAL EAR CAVITY. IN THIS CASE THE ARTIFICIAL EAR CAVITY IS USED ONLY AS A MEANS OF UNIFORM LOADING.

3. *Acoustic loading.* A loading or ultimate sound output limit test is conducted. This test is to determine the point at which the telephone ceases to increase acoustic output with increases of electrical input. In a well designed telephone, the magnetic saturation of the field will result before the armature strikes the pole pieces. The loading test determines this feature as well as the ultimate

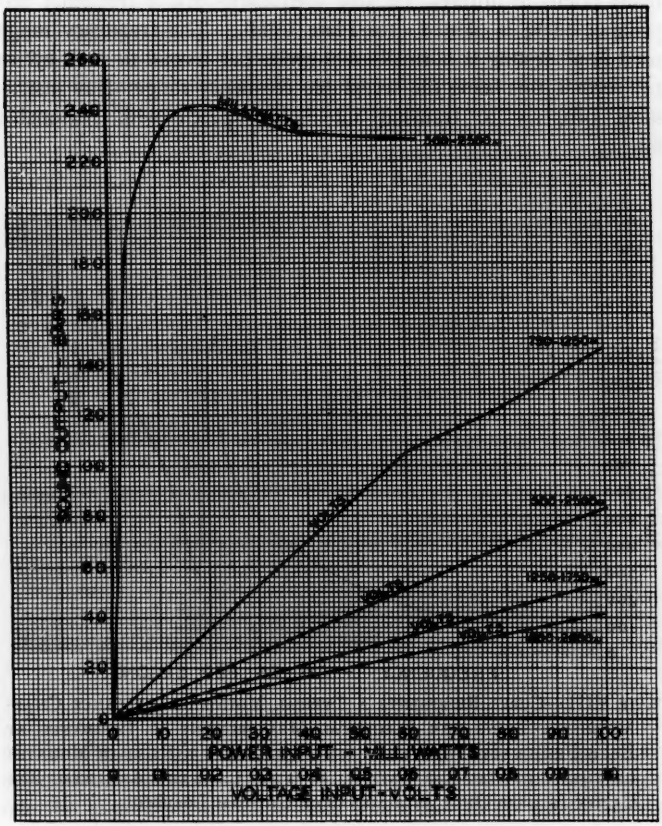

PLATE X.—THE CURVES REPRESENT ELECTRICAL EFFICIENCY OF A TYPICAL RECEIVER UNIT IN TERMS OF VOLTS INPUT AND MILLIWATTS INPUT FOR SOUND OUTPUT IN BARS.

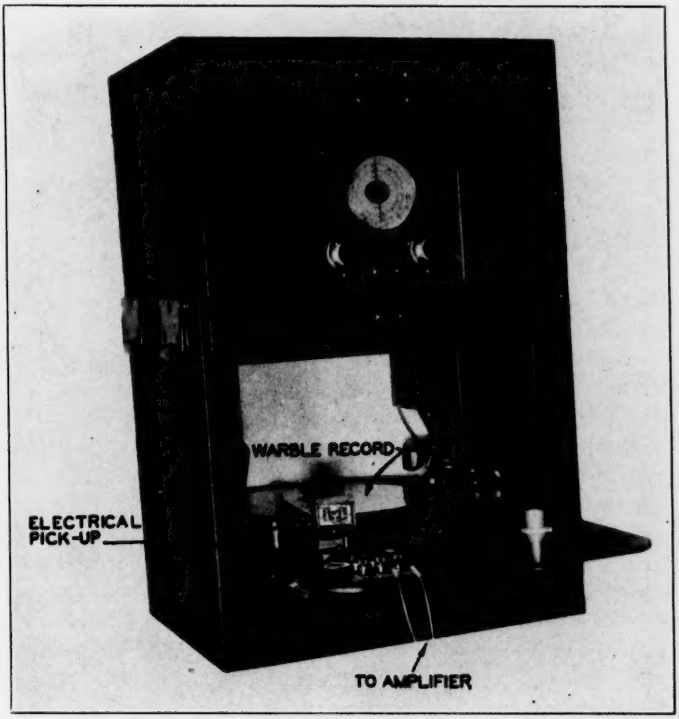

PLATE IX.—WARBLE FREQUENCY DEVICE FOR EXCITING TRANSMITTERS AT SELECTED FREQUENCY BANDS.

sound output which can be expected with unlimited electrical input. This is conducted by introducing an electrical warble frequency input from standard records (see Plate IX) through an amplifier, and increasing the average input to the telephone, recording the corresponding acoustic output on the response indicator. The data in this determination are computed on the basis of electrical voltage and acoustic pressure output, and the ultimate limit of the telephone's capacity is taken as the point where the plotted curve (see Plate X) falls off from a straight line, this indicating the point when acoustic output ceases to be linear with electrical power input. The taking of data is actually carried out to the region where the slope of the curve is zero. A well designed telephone should reach this saturation point at a relatively high acoustic output, with no evidence of the armature striking the pole pieces.

4. *Damping factor.* The damping factor, which is essentially the measure of the sharpness of the peaks in the telephone acoustic response is calculated from the frequency response curve. The unit is excited electrically in a circuit where two actual telephones are operated in series, the energy being supplied by the oscillator at a constant voltage across the system for all frequencies. In this case, one unit is mounted on the response indicator and its output measured, the second unit acting as line ballast suitably loaded. The two units in series are used, as in no case will an actual circuit comprise a single unit. The two units simulate the performance of the telephone in service. From the plotted acoustic frequency curve of the telephone on the response indicator, the damping factor is calculated. Excessively sharp peaks in the response of a telephone are of no advantage in service and indicate poor distribution of acoustic output over the frequency response range. Consequently, it is not considered to be an efficient acoustic instrument. The sharpness of the peaks in relation to their amplitude is definitely limited under Navy Specifications.

5. *Fidelity and intelligibility.* The fidelity, apparent loudness and intelligibility, are closely related to the usefulness of the telephone for speech reproduction and are all determined under test. The apparent loudness is determined by loading the telephone on the response indicator and driving it by electrical input comprising a series of warble frequency bands of constant amplitude. The

acoustic output of the telephone is measured on the response indicator, and the level of the response to each band plotted in dyne-centimeters units of acoustic output. The comparison of a combination of these bands (see Plate VII) with reference to their individual levels and their relation to one another, will show the apparent loudness of the telephone output. Loudness does not indicate a satisfactory telephone. A telephone may be loud but the intelligibility of speech may be poor. The loudness may be low and the intelligibility may be good, and in this type the telephone is useful in quiet spaces. However, intelligibility and fidelity, or the ability of a telephone to reproduce the speaker's voice in detail, must be combined, with a certain degree of loudness to produce a really satisfactory phone. Loudness is desirable, but only necessary as an accessory to fidelity and intelligibility. The test for the fidelity and intelligibility is done manually. The listener is seated in the beam of a volume controlled high power reproducer. A record of an aircraft motor is operated into the loud speaker, providing a general ground or masking noise approximating that in service condition. An announcer with average voice characteristics talks to the listener, both using the telephones under test. Short groups of words with no repetition are transmitted to the listener, and the masking noise level gradually raised, until a level is reached where he cannot clearly understand and repeat back word for word the message sent him. Upon the failure point being reached, noise level readings in decibels above a reference level of 0.000204 bar are made at his listening position and the value is a direct measure of the usefulness of the telephones for Navy service. It has been found that phones of medium loudness but high fidelity, are markedly superior under this test to a high loudness medium fidelity telephone, indicating that loudness without good fidelity is of less desirability than medium loudness with good fidelity. The two features "apparent loudness" and "fidelity" must therefore be analyzed before the grading of telephones can be completed.

6. *Determination of ruggedness.* Telephones are tested for ruggedness by mounting the units on a standard shock machine and subjecting them to a repeated shock applied indirectly to the telephone through the mounting of the machine. Before and after the telephones are thus shocked, they are subjected to the warble

frequency test to measure the apparent loudness. Any deterioration or change in their performance caused by the shock can be noted. In this measurement, the response indicator and speech generator are used, the former for measuring the receiver output and the latter for properly exciting the transmitter. Both transmitters and receivers are subjected to this shock and the testing is done in pairs.

7. *Corrosion resistance.* Other mechanical tests for material are conducted, such as the subjection to salt spray and the effects noted. The effect of wide and rapid temperature changes, and the general character of the design, construction and material is investigated. The purpose of these tests is to obtain materials which will withstand sea service for a long period of time.

By analyzing the results obtained from the foregoing tests there is available a method whereby accurate and definite comparative performance values may be assigned to the various types of telephones. In addition, points of faulty design or material weakness may be readily determined, and from the basis of test data, improvements are made to insure operation performance in excess of the minimum acceptable limits. Of even greater importance, however, is the fact that there is permitted a method of comparative test wherein the personal element is reduced to a minimum.

NOTES.

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British Motor Ship, January, 1936.

STEAM AND MOTOR SHIP COMPARISONS AGAIN.
British Motor Ship, January, 1936.

AMERICA'S WEAKNESS UPON THE SEAS.
Marine Journal, January 15, 1936.

AMERICAN SEA POWER AND WORLD EVENTS.
Marine Journal, March 15, 1936.

THE NECESSITY FOR THE REPLACEMENT OF SHIPS.
Proceedings of the American Merchant Marine Conference, 1935.

THE SECOND LONDON NAVAL CONFERENCE.
Journal of Royal United Service Institution, February, 1936.

THE NEW NAVAL TREATY.
Shipbuilding and Shipping Record, March 26, 1936.

FOREIGN NAVY NOTES.

RECENT DEVELOPMENTS IN HYDRAULIC COUPLINGS.
Engineering, May 3, 1935.

THE FUTURE OF STEAM PROPULSION.

There are few subjects of more enduring interest than that relating to the propulsion of ships, and so rapid have been the changes and progressive improvements in marine engineering technique during recent years that it would indeed be surprising were the position otherwise. As the birthplace and cradle of the steam engine and the home of engineers of the caliber of James Watt, Charles Parsons and Andrew Laing, this country has from the earliest days won for itself a pre-eminent position in steam engineering. This circumstance alone rendered it peculiarly appropriate that for the eighth Thomas Lowe Gray Lecture the chosen subject should have been that indicated by the title above, and in securing for last month's lecture a marine engineer so widely experienced and eminently qualified as Mr. John Johnson, the Chief Superintendent Engineer of Canadian Pacific Steamships, Ltd., the Institution of Mechanical Engineers are indeed to be congratulated.

Mr. Johnson, as one of the most courageous and progressive marine engineers of the present generation, needs no introduction to our readers. His valued contributions towards improved steam machinery performance, and particularly his successful pioneering efforts in the field of high-pressure steam, are well known far beyond the boundaries of this country, and the highly efficient Canadian Pacific fleet stands today as a monument to his enlightened and progressive technical policy.

At the outset, the lecturer indicated that the various forms of propulsion available today differed widely with respect to their cost and to their thermal and mechanical efficiencies, as well as in regard to their practical features; and it required great experience, judgment and foresight to arrive at the most profitable blend of these properties. Other desiderata were unfailing reliability, low maintenance charges, and freedom from noise and vibration; and the more nearly any system or combination approximated to these ideals, the more likely would it ultimately persist. Mr. Johnson then proceeded to examine on the one hand the case of the large express passenger liner carrying but little freight, and the 10-11 knot cargo tramp, on the other, and from these two types deductions were drawn applicable to intermediate-ship types. Conceding at once the great improvements made and being made in the Diesel engine, the author foresees the gradual discarding, one by one, of the cruder forms of steam engine, boiler and auxiliary plant, and the ultimate emergence and ascendancy of the geared steam turbine. The last-named, by reason of its demonstrated capacity to work at full designed power for a long term of years without depreciation and to use highly superheated steam without internal lubrication, must, in his view, ultimately displace all other forms of steam engine except for the smallest powers.

As was to be expected, Mr. Johnson produced some very interesting operating data from representative modern steam installations in the company's fleet with which he is associated. Illustrations were shown of the fire tubes and superheater tubes of the water-tube boilers of the *Empress of Japan* with a working pressure of 425 pounds per square inch and a temperature of 740 degrees F., and the absence of observable depreciation after five years' service was such that the author saw no reason why the tubes should not last the life of the ship. This excellent condition was ascribed to the closed-feed system, supplemented by a colloidal treatment to insure that the small amount of residual oxygen in the feed water was absorbed. With such treatment and given tight condensers, water-tube boilers can now safely be kept under steam continuously for six months, and have even been in continuous use for 12 months without detriment. Whereas in a large fleet boiler-scaling and cleaning at one time ran up an annual bill of £25,000, the figure today was less than £5000. It will perhaps come as a surprise to many who had re-

garded "condenseritis" as a thing of the past, to learn from the author that condenser-tube failures with 70-30 cupro-nickel still persist. The vulnerable area is confined to the first 2 or 3 inches from the tube inlets, signs of distress becoming apparent after five years' service in the tropics and eight years in the North Atlantic. Frank reference was also made to a variety of turbine-reduction gearing troubles; but having regard to the very high standard of design and precision now attained in gear manufacture, we find it rather difficult to reconcile the apparent conflict between the author's statements that on the one hand some "nervousness" still exists in shipping circles concerning reduction gearing and, on the other hand, that the *Empress of Britain* gears after five years' service and about 500,000 miles steaming show no sign of wear.

As an acknowledged authority on modern steam generators, Mr. Johnson's observations on the power plant of the immediate future for installations of over 20,000 S.H.P. are interesting and very significant. The steam conditions proposed are 450-pound pressure and 850 degree F. at the high-pressure turbine receiver, a vacuum of 29 inches and a boiler efficiency of 90 per cent. With regard to steam pressure, the figure, according to recent Continental standards, may appear conservative; and in insisting upon the retention of at least one steam drum in the boiler as a reservoir for maneuvering conditions, he is, in our view, merely having proper regard to the vital consideration of unfailing reliability of the boiler plant in service. For such an installation complete designs have already been prepared; and the salient characteristics are given as £10 per S.H.P. initial cost, 10 cubic feet per S.H.P. space occupied, 13 S.H.P. per ton weight of propelling machinery, and 0.50 pound of oil per S.H.P. per hour fuel consumption for propulsion purposes. It is further significant of the author's balanced cautiousness born of unique practical experience that he indicates his readiness, but only after a few years' experience of these steam conditions, to consider an advance to 500 pounds pressure and 900 degree F. initial temperature, subject to suitable materials becoming available for superheater tubes and certain other parts. The basic fuel consumption to be anticipated from such conditions would be 0.48 pound per S.H.P. per hour for propulsion alone. Of mechanically-assisted boiler-circulating systems and intermediate reheating of steam by independently-fired superheaters or equivalent methods, he is inclined to be skeptical.

From the foregoing it will be abundantly clear that, for liner propulsion with powers of 20,000 S.H.P. and upwards, Mr. Johnson favors only the geared steam turbine. Turbo-electric drive, already successfully established, finds no place in the lecture, nor do any of the several combination systems of reciprocator and exhaust steam turbine. For a 20,000-S.H.P. installation, the author considers that a Diesel arrangement either of two-stroke cycle, double-acting, direct or geared drive, would be 500 tons heavier than the corresponding steam installation, but would show about 20 tons per day lower fuel consumption, so that there would be little to choose between the combined weights of machinery and bunkers for a 10,000-mile voyage. In these calculations the author has adopted the unusual method of assessing Diesel-engine fuel consumption by adding 0.04 pound to the basic rate to take account of the equivalent cost of lubricating oil "consumed." He suspects, in his own words, that this may in certain circles be regarded as an "unpardonable liberty," and in this we believe he will not be disappointed. Later, in dealing with cargo-carriers requiring 1500 to 2000 S.H.P. for propulsion at 10 to 11 knots, the same fuel-consumption adjustment is made for the Diesel engine, and for an actual 1500-S.H.P. installation with two-stroke cycle, double-acting engines the basic rate is taken at 0.43 pound—a figure which is hardly representative of the best modern practice. The equivalent steam installation would comprise a 300-pound pressure air-jacketed im-

proved-type tank boiler—presumably of Howden-Johnson type—working at 750 degrees F. superheat in association with a double-reduction geared turbine, which would show a specific fuel consumption of 0.60 pound of oil. For a 10,000-mile run, the combined weight of machinery and bunkers would be 660 tons for the steamship and 640 tons for the motorship. The steamship is regarded as being more economical than the motorship so long as the price of furnace oil is less than two-thirds of that of Diesel oil, and the conclusion is drawn that for such a steamship cheap coal is likely to prove more advantageous.

Mr. Johnson is so far fair to the Diesel engine as to concede that for smaller powers steam propulsion must inevitably decline, but he is less convincing in his advocacy of steam for powers between 1500 and 2000 S.H.P. Something more is surely required than his statement that so far no single steamship of this power has been expressly designed and built to show what steam can really do, in explanation of the definite drift of old-established Scottish and North-East Coast cargo-steamship owners to the highly efficient and remarkably economical standard motor tramp. His case for steam propulsion of express ships of the liner class, on the other hand, is so convincing as to be unanswerable.

The paper as a whole is one which we can commend to our readers, and it is likely to be the subject of lively discussion for a long time to come.—"The Shipbuilder and Marine Engine-Builder," February, 1936.

WORLD TANKER CONSTRUCTION.

A Total of 940,000 Tons D.W.C., Including Motor Ships of 901,000 Tons D.W.C.

The following list gives details of all the motor tankers which, so far as is known, are now on order, and the majority of them have already been laid down. The total is 76 ships of about 901,000 tons deadweight capacity. It is understood that there are five steam tankers being built of about 40,000 tons deadweight capacity, so that the tankers on order on January 1, 1936, totaled 81 of approximately 940,000 tons deadweight capacity.

Of the motor tankers, 36 of 397,000 tons deadweight capacity will sail under the British flag. Norway is responsible for 12 of 172,000 tons deadweight capacity and Germany for nine of 114,000 tons deadweight capacity. The other tankers are being built for owners in various countries, but not more than three are for any one country.

It may be assumed that approximately two-thirds of the tonnage in question, or 600,000 tons deadweight capacity, will be completed during the course of the present year, and the remaining 300,000 tons early in 1937.

This output should not affect the upward tendency of the oil-tanker market during the coming year, bearing in mind that a considerable proportion of the ships are being built to replace tonnage broken up, and that if a life of 15 years be considered normal, tankers of approximately 450,000 tons deadweight capacity are needed this year to replace such vessels. But more important is the fact that there are now steam-driven tankers afloat of over 5,000,000 tons gross; a very considerable proportion of these cannot compete against modern motor ships and must, sooner or later, be replaced by them. It may also be added—a fact that is not generally realized—that the tanker fleet today is approximately the same, in tonnage, as it was more than four years ago.

Motor tankers now on order.

Owners.	Builders.	Tonnage.
		D.W.C.
Anglo-Saxon Pet. Co.	Lithgows, Ltd.	12,100
Anglo-Saxon Pet. Co.	Netherland S.B. Co.	12,100
Anglo-Saxon Pet. Co.	Netherland S.B. Co.	12,100
Anglo-Saxon Pet. Co.	Harland and Wolff	9,250
Anglo-Saxon Pet. Co.	Harland and Wolff	9,250
Anglo-Saxon Pet. Co.	Swan, Hunter and Wigham Richardson	9,250
Anglo-Saxon Pet. Co.	Swan, Hunter and Wigham Richardson	9,100
Anglo-Saxon Pet. Co.	Wilton Fijenoord	9,250
Anglo-Saxon Pet. Co.	Wilton Fijenoord	9,250
Anglo-Saxon Pet. Co.	De Schelde S. B. and E. Co.	9,250
Anglo-Saxon Pet. Co.	Rotterdam D.D. Co.	9,250
Anglo-Saxon Pet. Co.	Cantieri Riuniti dell'Adriatico	9,250
Anglo-Saxon Pet. Co.	Odense Steel Shipyard	9,250
Anglo-Saxon Pet. Co.	Blythswood S.B. Co.	12,100
Anglo-Saxon Pet. Co.	Blythswood S.B. Co.	12,100
Anglo-Saxon Pet. Co.	Swan, Hunter and Wigham Richardson	9,100
Anglo-Saxon Pet. Co.	Cammell Laird and Co.	9,100
Anglo-Saxon Pet. Co.	Harland and Wolff	9,100
Anglo-Saxon Pet. Co.	Van der Giessen and Zonens	9,100
Anglo-Saxon Pet. Co.	Rotterdam D.D. Co.	9,100
Anglo-Saxon Pet. Co.	Wilton Fijenoord	9,100
Anglo-Saxon Pet. Co	Netherland S.B. Co.	12,100
Anglo-American Oil Co.	John Brown and Co.	10,200
British Tanker Co.	Swan, Hunter and Wigham Richardson	12,250
British Tanker Co.	Swan, Hunter and Wigham Richardson	12,250
British Tanker Co.	Cammell Laird and Co.	12,250
British Tanker Co.	Harland and Wolff	12,250
British Tanker Co.	Harland and Wolff	12,250
British Tanker Co.	Lithgows, Ltd.	12,250
Eagle Oil and Shipping Co.	Blythswood S.B. Co.	12,100
Eagle Oil and Shipping Co.	Blythswood S.B. Co.	12,100
Eagle Oil and Shipping Co.	Lithgows, Ltd.	12,100
Eagle Oil and Shipping Co.	Lithgows, Ltd.	12,100
United Africa Co.	Deutsche Werft	14,500
T. W. Tamplin and Co.	Deutsche Werft	15,500
T. W. Tamplin and Co.	Deutsche Werft	15,500
Camillo Eitzen (A/S Navalis)	Kockums Mek. Verk.	14,500
Lorentzens Rederi	Deutsche Werft	14,500
Skibs. A/S Nanset (Iver Bugge)	A. B. Götaverken	14,500
N. Chr. Evensen (A/S Lampas)	A. B. Götaverken	14,500
Odd Berg Skibs.	Eriksbergs Mek. Verk.	14,500
Ludv. G. Braathen	Kockums Mek. Verk.	14,500
Sigurd Herlofson	A. B. Götaverken	15,000
Th. Brovig	Bremer Vulkan	14,500
Skibs. Ringfonn (United Africa Co.)	Deutsche Werft	14,500
Fearnley and Eger (United Africa Co.)	Deutsche Werft	14,500
Skibs. A/S Asplund	Kockums Mek. Verk	12,500
Texas Co. A/S	Deutsche Werft	14,000
Van den Bergh's Margarine A.G.	Howaldtswerke	6,000
German American Pet. Co.	Fr. Krupp, A.G.	15,000
German American Pet. Co.	A. G. Weser (Deschimag)	15,000
German American Pet. Co.	Blohm and Voss	15,000
German American Pet. Co.	Schichau Yard	15,000
Rhenania-Ossag Mineralölwerke A.G.	Deutsche Werft	12,100
Rhenania-Ossag Mineralölwerke A.G.	Bremer Vulkan	12,100
Rhenania-Ossag Mineralölwerke A.G.	Howaldtswerke	9,250
Van den Bergh's Margarine A.G.	Deutsche Werft	14,500
A. P. Möller	Odense Steel Shipyard	14,500
A. P. Möller	Deutsche Werft	15,000
Danish	Odense Steel Shipyard	14,500
Soviet Republic	Nicolaieffski Yard	10,000
Soviet Republic	Sormovo Works	7,600
Soviet Republic	Sormovo Works	7,600
Yacimientos Petroliferos, Fiscales	Hansen and Puccini	4,000
Cia. Arrendataria del Mon. des Petroles	—	10,000
Cia. Arrendataria del Mon. des Petroles	—	10,000
Trelleborgs Ang. Nya A.B.	Deutsche Werft	14,500
Trelleborgs Ang. Nya A.B.	Deutsche Werft	1,220
Rederi A. B. Pagota	A. B. Gotaverken	12,500
Soc. Auxiliare de Transports	Ch. et at. de St. Nazaire	10,000
Cie. Navale des Petroles	At. et Ch. de France	21,300
Mitsubishi Shoji Kaisha	Mitsubishi Dockyard	7,300
Nippon Tanker Co.	Kawasaki Dockyard	13,000
Standard Oil Co.	Burmeister and Wain	15,000
Standard Oil Co.	Fr. Krupp A.G.	15,000
Phs. Van Ommeren	Odense Steel Shipyard	14,500

Total 76 ships of 901,000 tons d.w.c.

—' British Motor Ship," January, 1936.

STEAM AND MOTOR SHIP COMPARISONS AGAIN.

It is a little difficult to take at all seriously some of the out-and-out steam advocates. Apparently they do not even trouble to read information relating to motor ships when it is presented to them. An article recently appeared in the journal of the American Society of Naval Engineers, entitled "Propulsion Machinery for Naval and Auxiliary Ships," by J. H. King, and this was mainly confined to a comparison between mercantile cargo steam and motor ships. Mr. King is associated with a well-known boiler manufacturing concern.

His statement that "the progress in Diesel machinery has been such that it is today quite suitable and satisfactory for specific installations," is very generous to those who believe in oil-engine propulsion, as only 70 per cent of the ships ordered in the last year are motor ships. The number of "specific installations" appears to be somewhat large.

Mr. King disputes the claim that Diesel machinery requires less space than steam plant. He puts forward a design for a hypothetical 6000 S.H.P. freight-carrying ship, which, he states, has a machinery space of 45 feet. British shipowners must be curiously unaware, or unappreciative of, modern steam development, since one of the latest freighters of about the size mentioned, equipped with 5200 B.H.P. steam machinery, the *Mawari*, has a machinery space of 75 feet. Mr. King has been "unable to find a single motor ship that even approaches the compactness of this steam installation." The *Port Townsville*, with 9000 S.H.P. machinery, has an engine-room length of 65 feet.

Mr. King claims for his ship—the like of which has not yet been built—an overall fuel consumption of 0.59 pound per S.H.P.-hour.

The same claims were made for the liners *President Hoover* and *President Coolidge*, but in service they averaged 0.68 pound per S.H.P.-hour. The overall consumption of many motor cargo liners with machinery of 6000 S.H.P. is about 0.37 pound per S.H.P.-hour for all purposes.

A final statement of Mr. King baffles us. It is to the effect that "it is generally acknowledged in marine work that a Diesel-engined ship cannot maintain its rated horsepower over any considerable period of time. . . . On the other hand, a steam plant can, and frequently does, operate at its designed full power continuously."

It is curious that what should be "generally acknowledged" in America, where not a single cargo liner has been built for years, and relatively few motor cargo ships at any time, should be totally opposed to the experience of at least 90 per cent of European and Japanese shipowners, who have had hundreds of motor cargo vessels in operation for many years.—"British Motor Ship," January, 1936.

AMERICA'S WEAKNESS UPON THE SEAS.*

Events of the last generation have proved that it would always have been a sound national economic policy for America to have maintained an adequate merchant marine under her own flag. Over three-quarters of our foreign water-borne commerce was carried in American ships for the period 1783 to 1860. For the half century from 1865 to the World War, during the transition from sail to steam, American ships carried less than 15 per cent of our foreign trade, and for the last decade of that period, the amount dropped to ten per cent. America's participation in the international carry-

* From an address by Alfred H. Haag, Chief, Division of Shipping Research, U. S. Shipping Board Bureau, before the American Merchant Marine Conference at New York.

ing trade had been reduced to negligible proportions and we were depending upon foreign ships to transport 90 per cent of our foreign commerce. This was in keeping with the fallacious argument propounded by short-sighted economists to permit the ships to carry our trade that would carry it cheapest.

After making an allowance for the earnings of our Government fleet since the war, it is plainly evident that our weakness upon the seas has cost us from the time of the Civil War nearly seven billion dollars—or an amount ten times that which would have been required by a fair and uniform system of Government aid. Notwithstanding the tremendous sums expended, we are today without a strong merchant fleet in the foreign trade.

The entire ocean-going merchant marine of the United States, of vessels 2000 gross tons and upward, consist of 1950 vessels of 11,138,000 gross tons, of which 432 vessels, of 2,311,000 tons, are employed on the Great Lakes, while 228 vessels, of 1,345,000 tons, comprise the Government-owned fleet. There are 341 tankers, of 2,385,000 tons. Vessels employed in the domestic trade total 461, of 2,032,000 tons, and in the foreign trade 488, of 3,065,000 tons.

Let us now analyze the ship tonnage owned in the world and the position the United States occupies in the international carrying trade.

The iron and steel steam and motor vessels of 100 tons and upward total 63,141,000 gross tons, of which Great Britain and its Dominions have 20,186,000, the United States 11,969,000, Japan 3,991,000, Norway 3,935,000, Germany 3,688,000, France 2,982,000, Italy 2,822,000 and the Netherlands 2,553,000 tons. These figures, however, include types which do not engage in the carriage of goods and passengers. Eliminating vessels trading on the Great Lakes, tankers, miscellaneous types, all vessels of less than 2000 gross tons, vessels in the American domestic trade of 2000 gross tons and upward, and 1,345,000 tons in the Government inactive fleet, there remain 35,900,000 gross tons of vessels, 2000 gross tons and upward, for the carriage of goods and passengers in the international sea-borne carrying trade. Eight of the principal maritime nations own 29,633,000 gross tons of this total.

In tonnage these countries rank in the following order: Great Britain 13,205,000, the United States 3,065,000, Japan 2,969,000, Germany 2,740,000, France 2,250,000, Italy 2,132,000, the Netherlands 1,757,000 and Norway 1,515,000 tons.

Vessels owned by these countries with speeds of 12 knots and upward total nearly 18 million tons, of which Great Britain has 8,320,000, Germany 1,941,000, the United States 1,555,000, Japan 1,551,000, France 1,492,000, the Netherlands 1,340,000, Italy 1,119,000 and Norway 597,000 tons.

Vessels ten years of age and under of these countries total 8,423,000 tons. Great Britain has 4,321,000, Germany 920,000, Japan 637,000, the Netherlands 602,000, Italy 560,000, Norway 559,000, France 527,000 and the United States 297,000 tons.

These figures show that from the standpoint of tonnage Great Britain outranks us more than four to one, with Japan owning about the same as the United States.

In ships with speeds of 12 knots and over we stand third, with Great Britain outranking us better than 5 to 1, Germany owning nearly 400,000 tons more, and Japan practically on a parity with us.

In ships 10 years of age and under the United States stands at the foot of the list, outranked by Great Britain 14 to 1, by Germany 3 to 1, by Japan and the Netherlands 2 to 1, and by Italy, Norway and France over 1½ to 1.

The foregoing figures demonstrate, conclusively, the weakness, in both carrying power and efficiency, of that part of the American Merchant Marine which operates in the international trade. Such deficiencies exist also in our domestic trade and tanker fleet.

A further analysis shows that in our foreign trade fleet, of 3,065,000 tons, 10-knot ships predominate, and over 85 per cent of the entire tonnage is 13 years old or more.

In our tanker fleet, of 2,385,000 tons, 92 per cent comprise vessels of less than 12 knots. Over 88 per cent of the entire tanker tonnage is 13 years old or more.

These figures disclose the imperative necessity for prompt modernization of our entire merchant marine, of which nearly 89 per cent, or about 6,638,000 gross tons, is now 13 years old or more, and of which 5,287,000 tons are already over 15 years old.

This will require an annual replacement program amounting to 375,000 tons in the foreign trade fleet, 271,000 in the domestic trade fleet, and 302,000 in the tanker fleet, or a total of nearly 950,000 gross tons per annum during the next seven years, in order to replace by 1942 the 6,638,000 tons of vessels which will then be 20 years old or more.

While the foreign trade fleet is drawn upon most heavily for naval auxiliaries during times of national emergency, there are also heavy demands made on both domestic and tanker fleets for military use.

The figures just quoted illustrate the high percentage of old and slow ships in all three categories of vessels, and the great shortage of suitable types of vessels to serve as efficient naval auxiliaries.—"Marine Journal," January 15, 1936.

AMERICAN SEA POWER AND WORLD EVENTS.

By The Editor.

There appears to be greater assurance today that America will reach Sea Power maturity than at any previous period in our recent history. The danger lies in neglecting the upbuilding of an adequate commercial fleet while we devote our energies to the acquisition of naval vessels and bases. Obviously, Sea Power rests upon both of these factors and, if we forge ahead on one leg of the proposition while ignoring the other, we will emerge from our efforts with a lopsided and ineffective solution at a very critical time in history.

It should be axiomatic that events rather than conversational propaganda determine the defensive policies of the leading nations of the world. It is to these events that we are indebtd for what practically amounts to a complete reversal of our recent policy of studied and consistent neglect of our sea forces.

The world is now at war on many economic fronts and this is only the prelude to open hostilities. Vast combinations of economic and industrial forces are secretly crystallizing into formidable aggressive units bent upon the acquisition of trade opportunities, key positions in potential foreign trade areas and the retention of imperial territories or mandates over vast populations, with all that that implies in exploitation of national resources and commercial infiltration.

The secret agreements discovered to exist following the conclusion of the World War shocked the so-called civilized world. They are a matter of indisputable record. Who can say that, as a result of the universal trade war now raging that similar under-cover agreements have not been consummated among the leading European powers? Not only must we give serious consideration to these secret engagements between European nations but we cannot ignore the threatening conditions overshadowing the Orient where solemn treaties and undertakings have been scrapped unceremoniously

because they stood in the way of imperialistic ambitions for the acquisition of territory belonging to weaker nations unprepared to defend themselves against aggression.

In the presence of all this realignment of world forces bent upon predatory designs against both weak nations and competitors, those who seek to destroy or curtail American Sea Power are guilty of treason to the Republic and should be publicly investigated and denounced as dangerous enemies. The public prints are filled with their handiwork in the form of slurring references to proponents of preparedness, Admirals of the Fleet, the American sailor wearing the Navy blue, armament manufacturers and associated interests engaged upon creating and supplying the demands of preparedness. Books have been written and circulated calculated to throw great industries into public disrepute; a certain minister of the gospel born in England uses his pulpit to discourage American preparedness by denouncing all forms of national preparedness; Congressional committees are appointed to investigate and use their information to crucify and destroy the spirit of America to defend herself in a hostile world that refused to follow our altruistic attempt to bring about disarmament by example.

The billions lent abroad are being used to build up tremendous naval and military armaments while we whistle for our money. Our people are subjected to sneering incivility and contemptuous treatment, while right here in our own land thousands of voices are raised to keep us in a position of unpreparedness, instill the seeds of slavish acquiescence into our youth and destroy the very foundations upon which must rest the determination of a free people to defend themselves against all forms of foreign aggression.

As matters stand we are encircled with island naval bases dominated by great naval and military powers. With the greatest coast line in the world to defend, we have allowed our Navy to fall to third place and have, through treaty agreements, actually lost control of the number of ships we may or may not build. Other nations unceremoniously scrap treaties that interfere with their naval and military plans which are, necessarily, instantly altered by threatening events. Only after numerous futile conferences have we finally awakened to the serious trend of world events and have made a determined stand for the type of naval craft best suited to our needs.

But we continue to follow the dead forms of international diplomacy and negotiation which have everywhere broken down under the pressure of reality. The League of Nations is moribund and has all but petrified into a monument to international futility. One after another international conference has broken up in discord leaving in its wake an accumulation of irritations and hostilities that lay dormant before the conferences convened. Still, we are optimistic and listen to Siren voices luring us further and further into the international complexities and chaos of the Old World where wars have been hatched for thousands of years, and where it is hoped our resources and man power will be pledged as a bargaining counter on the checkerboard of Europe where each move is made to win something at our expense and leave us with the shadow instead of the substance.

This is not an exaggerated picture of world conditions; it is based upon a considered and honest analysis of the discouraging facts that crowd in upon us from every reliable output vouched for by trained observers many of whom are fervent advocates of a synthetic world of international unreality which they reluctantly admit is crumbling into dust and which all the tears and sentiment in the universe cannot save.

And so, we must build Sea Power or perish. To be weak now is to fall before those who are strong. Events have moved in such a way, contrary to our hopes and expectations, that Sea Power is a necessity for national

survival. We are far behind and must build hastily to fill the gap in our sea defenses which include not only naval vessels but also auxiliary merchant ships. The first step has been taken—nothing more. It takes three years to build a battleship. Who can predict with any assurance in the present state of affairs what will happen in that time? Yesterday we were rich in illusion and in confidence. We lived through the golden years of opportunity. Today we stagger under heavy burdens and must count our pennies. Domestic problems perplex us and the future is dark and uncertain. Yet, necessity brooks of no delay. As we built our wartime merchant fleet at enormous cost and in breathless haste, so must we now forge the protective weapons of defense under pressure.

The details of how we are to overcome our Sea Power handicap are of less moment than the bare fact that at last we have awakened to the emergency character of the crisis that must be met with courage and determination. We have laid down a realistic naval policy and the first steps have been taken to carry on with it. But only the barest beginning has been made on the long road we must travel to adequate Sea Power. The General Board of the Navy has formulated the plans. No other agency of our government is so well informed as to our needs. The technical details of ship construction, total tonnage required, what is vital and most urgently required to place our country in a defensive condition can safely be left to our experienced naval officers. But our Navy officers are mute. They are not permitted to combat the sinister propaganda that circulates night and day from thousands of jury-rigged rostrums and that creeps stealthily into remote sections of the land to weaken the American spirit, sow seeds of doubt and build up hostile sentiment against national preparedness. This duty falls upon those who are in a position to speak out in defense of their country which is seriously menaced from within by demagogues on the one hand and by solemn portentous frauds on the other, who are past masters in the art of marshalling spurious argument in support of their un-American defeatist thesis.

While those of us who can do so, must combat these defeatist enemies in our midst, an effort should also be made to give the real facts to the people instead of distorted opinions based upon jumbled statistical outpourings designed to mislead public opinion. Neither our Navy or our merchant marine are as efficient as they should be. The tonnage and armament facts with regard to our naval establishment are so well known that I do not propose to enter here into a detailed description. Suffice it to say that we are far behind and will have to build day and night for several years to win back to the position we occupied in naval strength when our Sea Power was practically destroyed at the Washington Disarmament Conference in 1922 with the active aid of propaganda furnished by Lord Riddle who was unaccountably charged with the responsibility for giving out factual information regarding that disgraceful sell-out of American naval strength. Had the American public known the real facts, we would still be strong at sea. As it was they were hoodwinked and hornswaggled by misleading information with the active connivance of subservient American political bellwethers and stuffed shirts under the domination of England. Under Lord Hoover it was even worse, for the great engineer who trimmed the Chinese Mandarin, has been aptly described as " the best president England ever had in the White House."—" Marine Journal," March 15, 1936.

THE NECESSITY FOR THE REPLACEMENT OF SHIPS.

With the Effect a Building Program Will Have on Industry Generally to Bring the American Merchant Marine Up to Its Proper Status and Maintenance.

H. Gerrish Smith,
President, National Council of American Shipbuilders.

Mr. chairman and members of the Propeller Club and your guests: It was my privilege to be with you at your meeting in Savannah last year and to talk to you on a subject similar to that which is on my program for today. I stated at that time that the future of American shipping was clouded with a good deal of uncertainty, and I am sorry to say that the same situation exists today, although we have, of course, made a good deal of progress in some respects since last year.

I am particularly interested in appearing before you in the hope of getting a definite result from this meeting in New York. It is probable that every one of you within the reach of my voice is already pretty well sold to the idea that we need an American Merchant Marine and that we want to have it privately operated. I am wondering, however, whether we lose sight of the real reason behind this problem.

The real people who are interested in an American Merchant Marine are not here today. They may be here as operators of American ships or builders of American ships, but those who are interested from the broader national standpoint are those who have goods to sell abroad and whose markets will be increased and the safety in the carriage of whose goods will be enhanced by the existence and the operation of an American Merchant Marine. The American operator and the American builder are each of them simply contributors to the development of that transportation facility which is necessary to accomplish the above specific purpose.

PROBLEM OF NATIONAL IMPORTANCE.

It seems to me, therefore, that the particular problem before this conference is to get home in the broadest possible way to the American people the real facts involved, the fact that the problem is of national importance, and then to try to point out that we, who know the details, will do the best we can to get unified effort and coordination with a view to bringing about the proper result.

I think that the essay contest, conducted by the Propeller Club, if I may recite one little example, indicates the necessity for information concerning our American Merchant Marine and what it means to the people of the United States. I am told that one of the students who wanted to write an essay upon the subject, but who knew nothing about an American Merchant Marine, went to her teacher and asked, "What is an American Merchant Marine?"

After some deliberation the teacher said she wasn't quite sure, but she thought it was the fellow who pointed the gun aboard ship.

This particular individual then recited that she went to her father and to her grocer and to several other people before she could obtain any information regarding the merchant marine.

I think this little example illustrates the fact that not only in the writing of this essay did she enlighten herself to some degree and probably will enlighten others in the future, but that her teacher, her father and mother, and many others became somewhat familiar with what an American Merchant Marine really is and what it should be.

As a matter of fact, we have, as you know, a pretty respectable amount of American shipping on the seas in foreign trade. As far as the age of the ships is concerned, however, it is rapidly deteriorating. We are not keeping pace with the inevitable of either going forward or going backward at a very early date.

You know, all of you, what has been going on in Congress and in the various committees who have been investigating American shipping. You know the report of the committee that was dealing with mail contracts, and you have little confidence from the nature of that report that there is behind it a constructive program by the Committee for American Shipping. You know that efforts have been made to secure legislation in the past year. You know that those efforts have not been coordinated and are not unified.

REPUDIATION OF MAIL CONTRACTS.

I concur fully with the statement made by Senator Copeland in his letter read here today, that, whether you like it or not, you are faced with legislation for American shipping, that a "laissez faire" proposition will not do. If you go back into American history, you will find in 1858 Congress repudiated its mail contracts and ceased to render further aid to American shipping. It did the same thing in 1875 and, gentlemen, while I fully respect the validity of a contract I think that what Congress has done before, it is quite likely to do again, and I, for one, therefore, am satisfied that notwithstanding the fact that we have upon the statute books a very effective piece of legislation that might have given us a permanent policy and a permanent merchant marine, with perhaps only a modified requirement to take care of the cargo ship, we are now faced with a legislative problem, and I think it is up to us to get the best out of it that we can.

It is a case of getting legislation that is satisfactory and going ahead; if we don't, we are going backwards anyhow. That is my feeling, which is personal, of course, but I see no alternative except to follow along that line.

I don't mean the kind of legislation that was pending before the Senate when Congress adjourned at the last session, but I do mean legislation of a constructive character which is the kind that must be sought for and which I believe can be largely attained with unified action. This is no longer an individual problem. It is a national problem affecting this country more than it affects you, or me, or anyone in this room, or any group represented in this room.

INCENTIVE TO DEVELOPMENT.

I would look for legislation that would give an incentive to develop our present services, and legislation of a character that would leave the shipowner largely untrammeled in the operation of his ship, in order that he might make a profit, because it is hopelessly impossible to expect a continuation of our American services in foreign trade unless they can be operated at a profit. There will otherwise be no incentive to increase the services or to replace our obsolete ships, and the result will be nil.

I will now come back and see what the title of my subject is, and will give you a few points confined largely to a discussion of it. It reads, "The Necessity for the Replacement of Ships with the Effect That a Building Program Will Have Upon Industry Generally to Bring the American Merchant Marine Up to Its Proper Status and Maintenance."

In the development of this topic I have necessarily had to present a considerable number of statistics, and they are presented in chart form and those charts are made up separately for distribution, in the hope that they may be of use to your committee in transmitting the essential facts which they divulge to the American public.

CHART 1

GROSS TONNAGE OF SEAGOING MERCHANT VESSELS OF THE UNITED STATES

AS OF JANUARY 1, 1935

(Figures shown are for vessels each of 2,000 gross tons or over. The Shipping Board's permanent Idle Fleet of 1,358,582 gross tons and Great Lakes' vessels are excluded. Privately owned vessels temporarily laid-up are included).

COMBINATION PASSENGER & FREIGHT VESSELS & FREIGHTERS	Gross Tons
Foreign Trade	2,192,954
Coastwise Trade	1,914,130
Temporarily Laid-Up	1,010,713
Total:	5,117,797

TANKERS	
Foreign Trade	464,803
Coastwise Trade	1,786,739
Temporarily Laid-Up	138,721
Total:	2,390,263

TOTAL SEAGOING MERCHANT FLEET:	7,508,060

(Source: United States Shipping Board Bureau, Department of Commerce)

National Council of American Shipbuilders

I abhor statistics, but, living with them from day to day, I know that they are an essential part of the mechanics of business and they must be used, and in those that I have submitted I offer my apologies to Mr. Haag if at any point they conflict with the figures which he gave. I have used the figures which came from the Shipping Board, throughout, and have used them only in somewhat different form from that in which they were presented by Mr. Haag, so that upon analysis I think you will find they are in accord with his figures.

Now I refer you briefly to the various charts that are attached to my statement.

No. 1 shows a breakdown of seagoing merchant vessels by types and trades, classed as follows: Foreign Trade, Coastwise Trade, and Temporarily Laid Up. Freighters and passenger and combination ships in one group, and the tankers are in the other, indicating approximately five million gross tonnage of the first group and two million three hundred thousand of the other.

The charts speak for themselves, and I have presented those figures for the purpose of showing later on what the probable replacement program of ships for our seagoing vessels, for our coastwise and foreign trade will amount to, without taking into account the large number of smaller vessels that serve as their auxiliaries and that are employed upon our inland waters, gulfs and bays.

COMPARISON OF TONNAGE.

In Chart No. 2 I have shown United States tonnage as compared with the tonnage of other nations. As a matter of fact, we would be entitled to about twice as much tonnage in foreign trade as we actually have, if you based it upon the percentage that our foreign trade bears to the foreign trade of the world. I have always felt that we have been conservative in our demands as to the percentage of the carriage of our own goods in our own ships, and, as you have heard from other speakers, we are interested only to a very small degree in the carriage of goods for the other fellow. We either take what we sell to the other fellow or bring back what he has to sell to us.

Mr. Haag has emphasized the question of age and speed, and in considering the value of a commercial fleet to a nation, there must be taken into account not only the number of ships and tonnage in that fleet, but its age and the speed of the ships of which it is composed.

The United States ranks No. 2, with two very close competitors as far as the amount of tonnage is concerned, but when you come to the age of our ships we stand pretty near the bottom of the list in modern tonnage, and stand at the bottom as far as speed is concerned, so that to get a composite figure of the value of our fleet in comparison with that of other principal maritime nations you must take all those matters into account.

Chart No. 3 shows from the standpoint of speed, our relative position. We have 40 per cent of our total tonnage composed of passenger vessels each of 14 knots or over, and of combination vessels, and of freighters each of 12 knots or over, whereas every other nation has a considerably larger per cent of tonnage of these speeds or greater. Great Britain has 60 per cent. If you jump up another knot, say to 15-knot passenger vessels and to 13 knots for other types, the percentage of the United States is very, very much lower than 40, while other nations are more highly favored in that respect.

The charts show the six nations with which the comparisons were made. While the charts show the United States stands No. 2 in the amount of tonnage they also show that we stand last in the age of tonnage and in speed.

Chart No. 4 shows on a percentage basis when our ships will become obsolete. Going on an assumption pretty generally recognized the world over,

CHART 2

COMPARISON OF UNITED STATES SEAGOING MERCHANT FLEET

WITH THE MERCHANT FLEETS OF OTHER NATIONS

(EXCLUDING TANKERS)

AS OF JANUARY 1, 1935

(Figures shown are for vessels each of 2,000 gross tons or over. ALL TANKERS ARE EXCLUDED. Great Lakes vessels are excluded and in the case of the United States, the Shipping Board's permanent Idle Fleet of 1,358,582 gross tons is also exclueded).

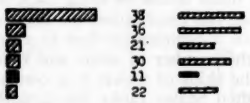
- Comparison According To Size -

Nation	Gross Tons
Great Britain	13,503,735
*UNITED STATES	5,117,797
Japan	3,014,308
Germany	2,694,438
France	2,249,118
Italy	2,150,491

(*) - Including Foreign and Coastwise Services.

- Comparison Of Gross Tonnage Less Than 10 Years Old -

Nation	Gross Tons	#Percent
Great Britain	5,102,176	38
Germany	965,736	36
Japan	644,256	21
Italy	642,593	30
UNITED STATES	556,473	11
France	495,873	22



(#) - Percentage figures indicate for each nation the percentage of its combination passenger and freight vessels, and freighters represented by tonnage less than 10 years old.

(Source: United States Shipping Board Bureau, Department of Commerce)

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CHART 3

PERCENTAGE OF COMBINATION PASSENGER AND FREIGHT VESSELS, AND FREIGHTERS REPRESENTED BY HIGH SPEED TONNAGE

AS OF JANUARY 1, 1935

(Figures shown are for vessels each of 2,000 gross tons and over. ALL TANKERS ARE EXCLUDED. Great Lakes vessels are excluded and in the case of the United States, the Shipping Board's permanent idle fleet of 1,358,582 gross tons is also excluded).

Nation	Total Gross Tons	*High Speed Tonnage	Percentage Of Total
Great Britain	13,503,735	8,084,405	60
Germany	2,634,438	1,612,313	60
France	2,249,118	1,148,011	51
Japan	3,014,308	1,426,819	47
Italy	2,190,491	911,377	43
UNITED STATES	5,117,737	2,030,171	40

(*) - High Speed Tonnage includes combination passenger and freight vessels each of 14 knots or over, and freighters each of 12 knots or over.

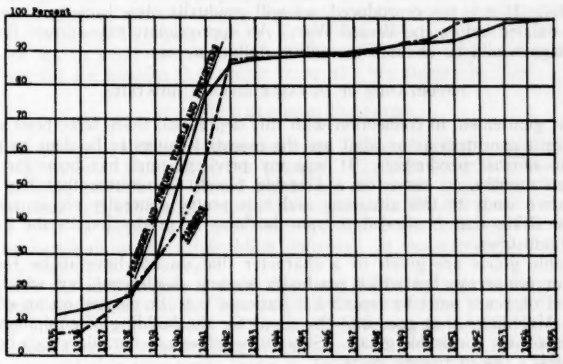
(Source: United States Shipping Board Bureau, Department of Commerce)

National Council of American Shipbuilders

CHART 4

PERCENTAGE OF UNITED STATES SEAGOING MERCHANT TONNAGE WHICH WILL BE 20 YEARS OR MORE OF AGE

AT THE END OF EACH YEAR FROM 1935 TO 1955 INCLUSIVE



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twenty years is the average life of a ship. While oil tankers are taken as having a life of twenty years, we know their effective life is somewhat less unless they are largely reconditioned.

This chart is very significant. Excluding tankers, our seagoing merchant fleet will be 30 per cent obsolete by the end of 1939, over 50 per cent by the end of 1940, 60 per cent obsolete at the end of 1941, and about 90 per cent obsolete in 1942, and the few which will not be obsolete will be mostly those constructed as a result of the Merchant Act of 1928 and will therefore form the only real nucleus of a modern American merchant marine.

Now, coming to this subject of replacement, I heard the statement made this morning by one of the speakers that with the type of legislation under consideration before the Senate when it adjourned, there wouldn't be any new ships built. I think he is about right, but I think our job is to try to get that legislation in such shape that new ships will be built, because this replacement is an inevitable phase of a permanent merchant marine program.

ANNUAL NECESSARY REPLACEMENT.

Our ships are so old they will soon be wholly out of the competitive class with their foreign competitors, and any program that does not take into account the question of replacement will be an utter failure.

In order to give some measure as to what this means, I have to use a few figures again. The active tonnage of our seagoing merchant fleet was never less than three million tons from 1926 to 1932, and it approximates the same amount of tonnage in the coastwise trade so that as a measure for future requirements figures are used about as they exist today. These figures seem to be conservative and, on a twenty-year basis, the normal replacement of this tonnage would require the annual construction of 150,000 tons of vessels for the foreign trade, and another 150,000 tons of vessels for the coastwise trade, making a total annual construction program of 300,000 tons of seagoing vessels.

Many people may say that such a construction program is ridiculous and cannot be expected, but to keep up our tonnage to what we now have on the seas over the average, that is the average tonnage that must be reproduced annually. If it is not reproduced, we will gradually slide back to a condition which existed before the World War. An approximate expenditure for such a tonnage would be about fifty million dollars a year.

IMPORTANCE OF DURABLE GOODS INDUSTRIES.

Now, gentlemen, in connection with this depression there have been studies by many committees as to what are the essential things to be done to get us back to normal production. It was my privilege, and has been for about eighteen months, to serve on a Durable Goods Committee that has given exhaustive study to this situation, and it is pretty generally recognized that the one thing that is needed to spur business is production in the durable goods industries.

Durable goods are goods of a character that do not have to be replaced today or tomorrow, but which gradually become obsolescent, but which over a period of years must be replaced if business is to be carried on an efficient basis. Machinery that goes into the shops for the building of equipment that goes into ships is durable goods. Ships themselves are durable goods because they have a life of twenty years.

An owner may wish to replace his ship. Conditions may not be right. He can wait a year or two years, but eventually it must be replaced. A spur to business that will bring about a replacement in the durable goods field is the

greatest spur that can be given to employment at the present time, because when durable goods start to improve, there is a general improvement, and there is nothing of which I know in the whole durable goods field that will give a broader spread of employment in industry than the building of ships because of the varied equipment that is required in their construction, and because of the very extensive distribution of the materials which go into the ships themselves, which is broadcast all over the land.

It has been estimated by the National Council of American Shipbuilders that during the lifetime of a vessel approximately 80 per cent goes to labor. In the shipyard itself all goes to labor. Outside the shipyard the vendor buys his materials, and in the production of those goods large sums go to labor, and, traced all the way, it is felt that 80 per cent is a conservative figure, and it has been consistently used for several years.

I mentioned a fifty-million-dollar program. We also are in possession of facts that show that during the lifetime of a ship for dockings, inspections, upkeep, changes, general modifications during its life, that from 50 per cent to three-quarters of the original cost is again expended over a twenty-year life. That means, using 50 per cent as a conservative figure, that on a fifty-million-dollar program there is another twenty-five millions for upkeep of a fleet of ships that in the long run is added to the actual money expenditure in the building of ships, without taking any account of the labor that is expended by the operator in the operation of the ships or in the purchase of supplies that are used in their operation.

Now, if you convert that seventy-five million dollars into labor, as illustrated on chart No. 5, you will find that in a breakdown of expenditures into items of material, shipyard labor, and that small group of expenditures that cannot be classified under either of the above heads, such as taxes, insurance, and miscellaneous items, it shows that the actual labor for the replacement of our present fleet of ships and upkeep, will give to American labor permanent annual employment to about forty-two thousand men; in other words, it will provide a livelihood permanently for at least two hundred thousand people a year, and, if you add to that the labor involved in the permanent operation of ships and into the production of the supplies which they use, that figure will be very largely increased.

That, gentlemen, without having read to you too many statistical details, gives the picture. The question is: Is it from the standpoint of labor employment worth while to promote facilities for the carrying of our foreign commerce for the benefits that accrue from it? My answer is, *Yes!*

AVAILABLE BUILDING FACILITIES.

I want to say at this point that there are plenty of facilities in the United States right now to build all of this tonnage. We have something like ninety-three building ways in the United States, not all of the same size, of course; some of them will build bigger ships than others, but 300,000 tons of merchant ships a year in addition to naval construction under way, will impose no building handicap on the shipbuilding industry. It means simply building up the personnel to do the work.

I might say further that if these ships are built they will in every respect equal the requirements of the International Convention for the Safety of Life at Sea, and in many respects will exceed those requirements. It is unfortunate that there is a prevailing impression that American vessels now in service are inferior as regards safety to those of other nations. This is not so. Notwithstanding the fact that the United States Congress has never ratified the International Convention, it was the United States that contributed largely to the high degree of safety prescribed by the Convention

CHART 5

LABOR EXPENDITURES INVOLVED IN ANNUAL EXPENDITURE OF $75,000,000

FOR THE BUILDING AND REPAIRING OF SEAGOING VESSELS

Diagram 1

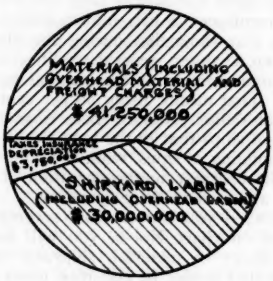

Diagram 2

Diagram 3

Diagram 4

National Council of American Shipbuilders

and the considerable number of ships built as a result of the Act of 1928 are all equal or in excess of the requirements of this International Convention.

At this point I want to commend the Propeller Club of the United States for the excellent work that I feel it did in the essay competition that it carried on, and I hope that it may be continued.

The geographical location of the several clubs comprising the members of the Propeller Club of the United States is such that whatever benefits may accrue to shipping as a result of this conference it could, it would seem, be brought home to a very large number of our people in the Propeller Club membership.

I thank you very much!—" American Merchant Marine Conference," 1935.

THE SECOND LONDON NAVAL CONFERENCE.

The Second Five-Power Naval Conference to be held in London opened on 9th December, 1935. This was convened with the object of drawing up a new agreement on the expiration of those arrived at during the First London Naval Conference in 1930, and at Washington in 1922, and before the end of 1936.

BRITISH PROPOSALS.

Mr. Baldwin, Prime Minister, in opening the Conference, said he would like to see a reduction in the size of all the larger types of ships and of the guns they carry. He still pressed for the abolition of submarines, failing which it was of vital importance to reach an agreement which would prevent their misuse.

UNITED STATES' PROPOSALS.

Mr. Norman Davis read a letter, written fourteen months previously, from President Roosevelt in which the latter suggested a proportional reduction of 20 per cent below the existing tonnage of the British, Japanese, and American Navies. Failing this or some less reduction, he proposed the maintenance and extension of existing treaties over as long a period as possible.

Mr. Davis admitted that in the interval the situation had changed considerably; but he advocated that the essential balance between the three Navies should be maintained by mutual agreement and not by extensive competition.

JAPANESE PROPOSALS.

Japan desired to see the abolition of all capital ships, aircraft carriers, and cruisers of the 8-inch gun class; but sought a considerable increase in her submarine strength.

She also wished to establish "a common upper limit of tonnage" for all those classes of warships which would be retained. This, in effect, was a demand for parity with Britain and the United States as regards "global" (total of all classes) tonnage, with freedom to apportion that tonnage to various classes as best suited her.

FRENCH PROPOSALS.

France favors drastic limitation both in maximum tonnage and caliber of guns. But M. Corbin, her spokesman, reverted to her old argument that, as regards quantitative limitation, the problem must be studied in relation to sea,

land, and air armaments. He implied that she was not willing to subscribe to the British idea of the total abolition of submarines, but emphasized that his Delegation would support any proposals calculated to humanize methods of naval warfare.

ITALIAN PROPOSALS.

Signor Grandi said that Italy had given eloquent evidence of her desire to limit naval armaments by having refrained from building up to her quota under the Washington Treaty. It was the principal aim of the Italian Delegation to avoid an armament race.

COURSE OF THE CONFERENCE.

With one exception, the proposals of the Five Powers are in time-honored form: Britain would like somewhat smaller—and therefore cheaper—ships, and more of them; Japan, France, and Italy would probably agree to an all-round reduction of size, but the United States' naval authorities are obsessed with the notion that the situation in the Pacific demands ships with a great range of action as well as speeds and armaments "second to none," and that these cannot be secured on lower displacements. The ever-recurring proposal to abolish submarines has met with no more support from France and Japan than it did when it was first made: it is regarded as purely self-interested on the part of Britain, and it seems unfortunate that it should be persistently put forward as a sop to sentimentalists in this country, knowing that there is no chance of it being accepted. France, ever with an eye on Germany and *securité*, dislikes these piecemeal discussions on armaments. She wants to be free to counter every move and development of her neighbor whether on land, sea, or in the air—all the more so now that the Treaty of Versailles is fast going by the board. Italy, in effect, re-echoes her prayer for "a navy as strong as France, but let it be a very little one."

The one striking exception in this Conference has been Japan's demand for parity in principle with Britain and the United States. This proposal met with such a frosty reception, especially on the part of the United States' delegates, that to avoid a premature *impasse* on 16th December, it was shelved "for further reflection and examination." But a month later the situation had to be faced that the Japanese Delegation had received imperative orders to adhere to this demand at all costs, as a matter of national prestige.

Lord Monsell, as British First Lord, pointed out that naval equality could not be measured in terms of ships, but questions of bases, supplies, and communications must also be taken into account. A Power with great naval needs could not denude its territories of their essential defenses and concentrate its entire fleet in the distant waters of a hypothetical enemy. The British Commonwealth had to take into account responsibilities in European waters, and in the Atlantic, Indian, and Pacific Oceans. Britain agreed with America that "it is impossible to maintain that equal armaments give equal security."

Mr. Norman Davies said that the United States could not consent to any change which would lessen their relative security.

Sir James Parr, speaking for New Zealand, remarked that the Japanese proposal would mean that the protection which the British fleet could give to his country would be almost negligible.

Mr. Bruce, on behalf of Australia, said that he saw in the Japanese proposal neither limitation nor equality of security.

Mr. Dulanty, representing the Irish Free State, supplied an element of, doubtless unconscious, humor. Having remarked at the opening meeting that

Saorstat Eireann had no navy and that the construction of vessels of war did not enter into the plans of his Government, he now put forward an emphatic demand that the Irish Free State should be regarded as a high-contracting Power and should be recognized as such in any agreement reached.

The outcome of this meeting was that Admiral Nagano, head of the Japanese Delegation, addressed a letter to the Vice-President Lord Monsell, intimating that the Japanese had withdrawn from the Conference. This was thereby reduced to a Four-Power status, Japanese representatives remaining as "observers."

FOUR-POWER TALKS.

The reduced Conference then proceeded to discuss three plans for the exchange of information submitted by the British, French, and Italian delegations, and it was agreed in principle that advance notification of building programs was a desirable feature of any agreement for limitation of naval armaments. The matter was referred to a sub-committee to draft proposals.

On 30th January, the Chairman having remarked that the Conference had agreed in principle to confine its discussions to practical rather than ideal proposals, and that there was little hope of securing agreement on qualitative limitation on the lines advocated by the British Government, then put forward certain figures as a basis for discussion. This proposal was agreed to by all the Delegates, and it was decided to refer the matter, in the first instance, to a technical committee. The figures are:—

Capital Ships.—35,000 tons and 14-inch guns, with a possible reduction of 2000 or 3000 tons in this displacement.

Note.—The existing limitations for this class are 35,000 tons and 16-inch guns. (Washington Treaty.)*

Aircraft Carriers.—22,000 tons and 6.1-inch guns.

Note.—The existing limitations for this class are 27,000 tons and 8-inch guns. (Washington Treaty.)†

A-Class Cruisers.—No more to be built during the term of the proposed agreement.

Note.—These are vessels of approximately 10,000 tons carrying 8-inch guns.

B-Class Cruisers.—7500 to 8000 tons with 6.1-inch guns.

Note.—Within the total quota of cruiser tonnage, "B" Class Cruisers can be and are being built up to 10,000 tons displacement.

Submarines.—2000 tons and 5.1-inch guns.

Note.—These limitations are the same as those agreed to by all Five Powers at the First London Naval Conference.

It would be premature to remark on the probable outcome of the Conference or its effect on our own Navy. It is obvious that any agreement which is likely to be come to must be far less comprehensive than those which resulted in the Washington and London Treaties. No arrangement for quantitative limitations can be satisfactory which does not include Japan. If the Four Powers can agree on qualitative limitations, there can be no doubt but that their example will exert a very considerable and probably a predominating influence on other Navies.

But above all else, it is to be devoutly hoped that we shall soon be free to build, and that we shall proceed to build, up to the needs of our own special naval requirements, and that we shall have shaken off, once and for all, such fetters as those which were placed on us by the First London Treaty.—"Journal of Royal United Service Institution," February, 1936.

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* H. M. S. *Hood* of 41,200 tons was excepted from this rule.

† Each of the Contracting Powers was permitted to build two carriers each of not more than 33,000 tons, hence the *Lexington* and *Saratoga* (U. S. A.) of this displacement.

THE NEW NAVAL TREATY.

FROM OUR NAVAL CORRESPONDENT.

The work of the 1936 Naval Limitation Conference has resulted in the signature of a Treaty by three of the five Powers invited to the discussions; in the promise by a fourth Power to sign later; in the possibility (but no more) that the fifth may "adhere" to the provisions; in the negotiation of a separate treaty between Britain and Germany which will bind the Reich to all the restrictions accepted by the other Powers; and in the expectation that Russia will also conform to the limitations.

Such a summary of the results gives, however, but a partial and, perhaps, an unduly rosy view of the position. For the amount of limitation actually imposed by the clauses of the Treaty is far less than was hoped for when the Conference assembled, and in practice will prove to be very little more than would have been imposed by the natural circumstances, financial and technical, in which the replacement programs of the next few years will be carried out.

Limitation of the number of ships in each class in each fleet has not been achieved. That is the really regrettable feature of the Conference, though it must be admitted that few of those who had given any study to the subject previously could see any way in which the varying objections of the different Powers to perpetuation of the Washington ratios were to be overcome and all brought to a mathematical agreement. The only possibility of quantitative limitation lies now in the new system of interchange of information about programs and the machinery for discussion among the Powers, prior to the laying down of keels, of any unduly large programs.

This is something entirely new in international relations and the application of the system in practice should prove extremely interesting. The secrecy which has hitherto shrouded programs and designs is to be abandoned, and the signatory Powers are annually to exchange detailed information under both heads four months before any ship is laid down. It must be noted that Japan has not yet given her adherence to the project, and in view of the deep secrecy with which she has hitherto surrounded all her warship building activities, it is feared that she may decide to retain her freedom of action. Will that necessarily wreck the whole scheme? In Conference circles the tendency has been to consider that Japan's abstention would not be a serious obstacle. If six Powers are fully informed of each other's potential strength at sea the projects of a seventh Navy have less influence on the situation than at a time when all are working in the dark.

The qualitative limitations accepted under the Treaty are set far higher than most of the Powers desire. Battleship design could easily have been limited to 27,500 tons, but for the immovable opposition of the United States technicians. The most that the other delegations have been able to achieve here is an agreement that the whole question shall be reviewed in three years' time. The "tin-clad" cruiser which was born out of the Washington Treaty is shelved, probably for all time, by the agreement that no more 10,000-ton vessels shall be built during the six years that the Treaty is in force. The British delegation's desire to see severe limitation imposed on the size of torpedo craft and submarines has been disappointed. None of the other Powers sympathized with the idea, and in the result the Treaty is so framed that no restrictions on the size of destroyers are laid down while submarines are allowed to go as high as 2000 tons. We have seen one effect of this already in the orders for the new British "tribal" class destroyers of 1850 tons.

No new problems so far as the Great Powers are concerned have been posed for the naval architect by the agreement. Those who are concerned with the minor navies, however, are confronted by the curious stipulation that no ship shall be built between 8000 and 17,500 tons. These are the limits within which most of the "capital ships" of the smaller fleets are designed, and while it is true that the agreement does not bind any but the big Powers, the evolution of a new and effective type within this "non-construction zone" for a minor Navy might have a serious effect on the whole agreement. The amount of discussion in private at the Conference about the place in the future categories of the German pocket battleships was an indication of the possibilities.—"Shipbuilding and Shipping Record," March 26, 1936.

FOREIGN NAVY NOTES.

ENGLAND.

The following vessels were launched in 1935:

Cruisers.—Ajax, Arethusa, Galatea, Sydney (for R. A. N.).

Flotilla Leader.—Faulknor.

Destroyers.—Fortune, Fame, Forester, Firedrake, Foresight, Foxhound, Fury. (N. B.—*Fearless,* of this class, was commissioned on 22nd December, 1934.)

Submarines.—Salmon, Snapper, Clyde.

Sloops.—Enchantress (late *Bittern*), *Londonderry, Wellington, Deptford, Indus* (for R. I. N.), *Yarra* (for R. A. N.).

Coastal Sloop.—Kingfisher.

Minesweeping Sloops.—Hussar, Speedwell.

During 1936 the following ships should be completed:—

Cruisers.—Amphion, and *Apollo* of the 1932 program; and *Penelope* of the 1933 program.

Flotilla Leaders.—Grenville (1933) and *Hardy* (1934).

Destroyers.—Greyhound, Griffin, Garland, Gallant, Gipsy, Grenade, Grafton, Glowworm (1933), *Hero, Havock, Hasty, Hunter, Hereward, Hotspur, Hostile, Hyperion* (1934).

Submarines.—Grampus and *Narwhal* (minelaying submarines), and *Seawolf* (small coast defense type), all of the 1933 program.

Sloops.—Aberdeen, Fleetwood and *Swan* (for R. A. N.).

Minesweeping Sloops.—Niger, Salamander.

1935 Program.—Names have now been allotted to the ships of the 1935 building program:—

Cruisers.—all of the *Southampton* class (9000 tons)—are to be called *Liverpool, Manchester* and *Gloucester. Manchester,* like *Sheffield,* of the 1934 program, is a new name to the Navy List.

Submarine Depot Ship.—Maidstone.

Flotilla Leader.—Inglefield.

Destroyers.—Icarus, Ilex, Imogen, Imperial, Intrepid, Impulsive, Isis, Ivanhoe.

Submarines.—Cachalot (minelayer), *Sterlet* (Swordfish class, coast defense submarines) and *Triton.*

Minesweeping Sloops.—Hazard, Hebe, Sharpshooter.

Convoy Sloop.—Bittern.

Coastal Sloops.—Sheldrake, Kittiwake.

Surveying Ship.—Gleaner.

Magnetic Survey Ship.—Research.

Coastal Minelayer.—Plover.

Boom Gate Vessels.—Ebgate, Bargate.

1934 Program.—All ships of this program are now building.

1933 Program.—The dates of launching of the two remaining cruisers of this program are *Newcastle,* on 23rd January, and *Southampton,* on 10th March.

All the destroyers (G Class) were launched during 1935, and should be ready for service during the present quarter.

If the practice of recent years of not laying down ships during the year in which they appear in the Naval Estimates is maintained, it is probable that the *Liverpool, Manchester,* and *Gloucester* will be the last to be built under the restrictions of the present London Naval Treaty, which expires at the end of the year.

Battle Cruisers.—The battle cruiser *Repulse,* after being in dockyard hands since 1932, for large repairs, commissioned in January for service. On 29th October the Admiralty announced that an order for new machinery for H. M. S. *Renown* had been placed with Messrs. Cammell Laird, Birkenhead.

New Coastal Motor Boats.—The Admiralty have ordered from the British Power Boat Company, Southampton, six coastal motor boats. This is the first order for this type of craft since 1922.

Speed Boats for Destroyers.—Trials have recently been carried out at Southampton of high-speed launches for use by destroyers. Twenty-five of these boats have been ordered. Their length is 25 feet. They have a 100-horsepower engine which gives a normal speed of 22 knots and a maximum of 26 knots. They are stated to be very handy craft and remarkably seaworthy.

FRANCE.

New Battleships.—The two 35,000-ton battleships will be named *France* and *Patrie.* The former has been laid down in the Salou basin at Brest. The latter will take the place of the *Strasbourg* on the slips at St. Nazaire. The *France,* like the *Dunkerque,* will have the bow portion built separately.

Modernization.—The battleships *Provence, Lorraine,* and *Bretagne* have all been modernized. In the case of the *Lorraine* the center turret has been removed and replaced by a catapult with stowage for two aircraft. The protection of all three ships has been improved, the elevation of the main armament increased, and the boilers are now oil fired. It is possible that these ships may now be capable of 23 knots.

New Cruisers.—The fifth of the *B*-class cruisers, *Montcalm,* was launched on the 20th October. The whole of this class should be in service before the end of 1937.

Submarines.—France now has the greatest submarine flotilla in the world; built or building, she possesses 111 underwater craft. The largest is the *Surcouf,* of 2880 tons, carrying two 8-inch guns and 14 tubes. The smallest are craft of 597 tons, which carry one 3-inch gun and eight tubes. The latest first-class submarines have a surface displacement of 1379 tons and carry one 3.9-inch gun and 11 tubes.

Cruise of the 2nd Squadron.—A cruise of the 2nd Squadron from Brest commenced on the 16th January. The ships are visiting ports on the North-West coast of Morocco and are carrying out exercises. They were due to return to Casablanca on the 11th February, and to Brest by the 26th February.

It has been formally denied that these movements have anything to do with the international situation, or that it was the intention to reinforce the 1st Squadron in the Mediterranean while the question of oil sanctions was being discussed at Geneva.

GERMANY.

Admiralty.—The *Marineleitung* has been considerably increased to meet the expansion of the Navy. The Commander-in-Chief of the Navy, Admiral Raeder, has a Chief of the Staff, Admiral Densch, a personal A.D.C., two liaison officers for the Naval Attachés, a communication officer, and a Surgeon Commander, and there is a Naval A.D.C. to the Chancellor, who also works in the C.-in-C.s office. The six Naval Attachés are directly under the Commander-in-Chief. The Personnel Section is under a Kapitan zur See, who now has a staff of twelve. The Naval Staff Departments have been increased from 43 to 93, and the Construction, Equipment, Hydrographic, Navigation, and Dockyard Departments *(Allgemeine Marineamt)* from 42 to 73. The Ordnance Department is under a Rear-Admiral. There is also a large number of naval officers in the *Wehrmachtamt* (part of the War Ministry), also in the *Wehrwirtschaftliche-Abteilung* (Department for Industrial Mobilization).

German Fleet.—It is understood that the new battleship *Admiral Graf Spee* will be the Fleet Flagship of the Commander-in-Chief. The flag of the Admiral commanding the Battle Fleet is flown in the *Deutschland*. The *Leipzig* is due to relieve the *Königsberg* as flagship of the Scouting Forces.

Naval Bases, Etc.—It is understood that Kiel will in future be the main battleship base, and the dry docks there are to be enlarged to take the new ships. A floating dock is also being provided. The cruisers will use Wilhelmshaven. Emden, Bremerhaven, Swinemunde, and Pillau will have an increased number of vessels based on them.

Inspectorates.—Under the new organization there are Inspectors of Training, Torpedoes and Mines, Gunnery, and Engineering. The first three are each under a Flag Officer, and the latter under an Engineer Rear-Admiral.

ITALY.

Battleships.—It is reported that the new 35,000-ton battleships *Littorio* and *Vittorio Veneto* will be 754 feet long, 103⅔ feet beam, and approximately 30 feet draught. An armament of nine 15-inch guns in triple turrets and a speed of 34 knots have also been mentioned. In addition, Italy has four older battleships, two of which, the *Cavour* and *Cesare,* are undergoing extensive reconstruction.

Cruisers.—There are twenty-seven cruisers, and three more building. Of the latter the *Savoia,* of the so-called *Condottieri* class, although nominally of only 6791 tons, is estimated to displace approximately 9000 tons with stores and ammunition on board. Two later ships, the *Abruzzi* and *Garibaldi,* will be armed with ten instead of eight 6-inch guns, and will certainly displace at least 9000 tons instead of the scheduled 7874; the former was due for launching early in 1936, and the latter probably later in the year.

Destroyers.—There are ninety-eight leaders, destroyers, and torpedo-boats; and eight more building. A recent addition to the building program has been four vessels of 1850 tons laid down at the Orlando Yard. These will carry four 4.7-inch and eight smaller guns and six tubes.

Submarines.—Sixty-four submarines are also with the fleet, and thirteen more are building.

JAPAN.

Capital Ships.—All the capital ships should have been modernized by the end of the fiscal year 1936-37. Some of them have already been in hand twice for this process: the *Karishima* and *Kongo,* originally battle cruisers, are now

undergoing a second treatment. The *Ise* and *Hyuga* will be the last to be dealt with. They are the only battleships which still burn coal, and will doubtless also be converted to oil.

Aircraft Carriers.—The 10,000-ton aircraft carrier *Soryu (Blue Sky Dragon)* and a 10,000-ton seaplane carrier, the *Chitose,* are both building at Kure. Another aircraft carrier and two seaplane carriers are projected.

Cruisers.—Three of the new 8500-ton cruisers, the *Mogami, Mikuma* and *Suzuya,* are now complete. The *Kumano* has been launched, and the *Tone* and *Chikuma* are on the stocks. These ships carry fifteen 6.1-inch guns, eight 5-inch A.A. guns in twin turrets, and twelve torpedo tubes. They are fitted with catapults for two aircraft. The *Tone* will have an increase in beam as compared with the earlier ships, possibly in order to give better stability.

Destroyers.—With a displacement of only 1368 tons, Japan is mounting an armament of five 5-inch guns and eight torpedo tubes. This is almost equal to the armament of French torpedo craft of nearly 1000 tons greater displacement; but the designed speed of the Japanese vessels is only 34 knots as compared with 37 knots in the latter, which in practice have greatly exceeded that speed on trials, running light.

Six of the twelve destroyers of the *Ariake* class are complete. The single 5-inch superimposed gun has been removed from the original position before the bridge and is now mounted back to back with the twin turret aft. This alteration follows the capsizing of the torpedo boat *Tomodzuru* and has doubtless been done to reduce top weight.

Torpedo Boats.—Sixteen torpedo boats of 395 tons, small editions of the *Tomodzuru* class, are building or projected. They will have an armament of three 4.7-inch guns and three torpedo tubes. The designed speed is 28 knots.

Submarines.—In addition to the three normal classes of submarines—cruiser, fleet, and coastal types—Japan is building miniature submarines. It is understood that these are craft displacing only about 12 tons, with a length of 32.5 feet, electrically driven and having a very low under-water speed. The armament consists of one torpedo tube and a machine gun. There is a crew of four, and the vessel can remain submerged for about three hours.

RECENT DEVELOPMENTS IN HYDRAULIC COUPLINGS.*

By Harold Sinclair.

Broadly, there are two classes of hydraulic power transmitters:—

(1) The hydro-kinetic or turbo type, in which the power is transmitted by the kinetic energy of a liquid discharged by an impeller on the primary shaft against the vanes of a turbine runner on the secondary shaft. Examples are the Föttinger transformer, the Vulcan hydraulic coupling, the Vulcan-Sinclair coupling, the Vickers-Coats torque converter, the Lysholm-Smith torque converter, the Voith turbo-converter, and the Voith-Sinclair turbo-coupling. The name "hydro-kinetic" is suggested as being most descriptive of the principle involved, since the torque transmitted is dependent upon the velocity of movement of the working medium.

(2) The hydro-static or displacement type, in which the power is transmitted by liquid delivered under pressure by a positive variable displacement pump to a constant-displacement hydraulic motor, usually of the piston or eccentric-vane type. Examples are: the Lentz, the Williams-Janney, the Hele-Shaw, the Lauf-Thoma, the Schwartzkopf-Huwiler, also the Hele-Shaw

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hydraulic clutch and the Krupp clutch. The term "hydro-static" is not a very happy one, but is suggested as being the most descriptive of the principle involved, having regard to the fact that the torque transmitted is proportional to the pressure and is independent of the velocity of the working medium. Both classes of hydraulic transmissions include examples of: (*a*) torque converters or gears, in which the output torque varies, or can be controlled, inversely as the speed; and (*b*) couplings or clutches, in which the output torque is equal to the input torque and is usually provided with some degree of regulation of the slip.

Overheating of a displacement type gear must be carefully guarded against, whereas with a turbo type transmission the working conditions are actually improved by high-temperature operation, provided the temperature is kept within the limit at which continuous working will not cause deterioration of the oil. Owing to the positive character of the displacement type of transmission, its characteristics are strikingly different from those of turbo transmissions, and there is a considerable range of applications for which hydrostatic gears are ideal, but for which turbo transmissions have little or no scope. In particular, one may mention all applications of low or medium power requiring precise speed regulation over a wide range of torque and speed, and a constant speed of the output shaft practically irrespective of changes in load. Turbo transmissions are not generally suited for such purposes, on account of the turbine characteristic that the speed is responsive to changes in the load. It is this feature, incidentally, that permits of the automatic variation of torque with speed that is so useful in the torque converter. Where the load fluctuations on a turbo-coupling of the variable filling type are not very rapid, such speed changes can be corrected by governor gear, but the natural characteristic is broadly similar to that of a shunt-wound direct-current motor, or a slip-ring alternating-current motor, with a variable resistance in the rotor circuit. The turbo-coupling comprises merely a primary shaft, with its impeller, and a secondary shaft, with its turbine runner; there are no fine running clearances, heavily-loaded bearings, or parts affected by distortion.

There is a wide field of application in which either the hydro-kinetic or hydro-static transmission can be utilized. As regards variable-speed drives, the efficiency of the hydraulic coupling with variable filling or slip control is definitely superior at the high-speed end of the range, but falls well below that of the hydro-static type when working at a large speed reduction. In the case of fan or centrifugal pump drives this is unimportant, since the load falls off rapidly with reduction in speed and the actual energy loss in the transmission is small. On the other hand, for constant-torque drives the hydraulic coupling offers the convenience of infinitely-variable speed regulation, but gives no advantage in saving of power when running at reduced speeds. The hydraulic coupling offers further advantages in the shape of light-load starting, limited stalling torque, and flexibility of connection in both the lateral and torsional senses, but its outstanding feature is mechanical simplicity and smooth vibrationless drive. It appears to the author that the latter reasons explain the very wide extent to which hydro-kinetic transmissions are now being used, marine developments having accounted within the last eight years for over 750,000 horsepower, and automotive and general industrial drives for a further 500,000 horsepower during the last four years.

There is a good deal of confusion concerning the fundamental differences between fluid couplings and fluid torque converters, and the idea is quite general that a fluid coupling is capable of at least some degree of torque conversion when working at a high torque and slip. Hence it should be emphasized that the paper deals solely with hydraulic couplings, which, although

incapable of torque conversion, are in fact used for a much wider range of applications than torque converters. Fluid couplings are essentially of the two-element type having only a primary member and a secondary member, without any intermediate guide vane or reaction member such as is necessary to obtain torque conversion. The torque at the output shaft is equal to the torque at the input shaft under all conditions of speed, filling, and slip. Torque converters, on the other hand, have essentially three elements, an impeller on the input shaft, a turbine runner on the output shaft, and a stationary guide vane or reaction member interposed in the fluid stream circulating between the primary and secondary members, and serving to change the direction of the stream of liquid in such a way as to alter the torque ratio between the input and the output shafts. The turbine and the guide-vane elements may be divided into two or more stages, and the vanes may be adjustable or movable, as in the case of some of the early Föttinger speed transformers, the Vickers-Coats gear, and some designs of Lysholm-Smith torque converter. The guide vanes may be curved sufficiently to reverse the direction of the fluid stream, in which case the turbine runner is arranged to suit, and the transmitter becomes a reversing gear.

Historical.—The first hydro-kinetic transmission was that originated by Dr. Föttinger when on the engineering staff of the Vulcan-Werke A. G. of Hamburg and Stettin, prior to the war. At that date the steam turbine, which had been newly introduced for marine propulsion, had to be directly coupled to the propeller shaft; as a result the turbine speed was too low for efficient operation, while the propeller speed was too high, having regard to the different densities of the media in which the turbine and propeller were working. Dr. Föttinger conceived the idea of a hydraulic speed and torque transformer or gear comprising a centrifugal pump combined in a single unit with a turbine runner, giving a speed reduction ratio of the order of 5 to 1 and thus permitting the steam turbine and propeller to work under conditions so much more efficient as to compensate for the hydraulic losses of 15 per cent or so in the transmission.

Föttinger speed transformers were made by the Vulcan-Werke in sizes up to 10,000 and 15,000 horsepower and for a time enjoyed considerable success, so that a total construction of about 220,000 horsepower was reached. The introduction by Sir Charles Parsons of helical reduction gearing for turbine drives, however, put an end to the marine application of the Föttinger transformer, owing to the much higher efficiency and lower cost of the mechanical gear, and its compactness and the availability of large reduction ratios.

For some time after the war the development of fluid transmission remained at a standstill until Dr. Bauer, the well-known marine engineer and director of the Vulcan-Werke of Hamburg foresaw the use of high-speed geared Diesel engines for marine propulsion in place of the large slow-speed direct-coupled engines which were then standard practice. A great deal of money and energy had been expended upon the perfecting of the Föttinger transformer to meet the competition from mechanical gearing, and the most exhaustive tests and analyses had been carried out to ascertain the nature and amount of the individual hydraulic losses. Efficiencies of the order of 85-87 per cent were realized, and it is believed that this represents nearly the maximum possible for such a transmission, as it does not appear to have been surpassed since. It was established that the major losses due to fiuid friction are inseparable from the existence of the stationary guide vane reaction member which is essential in a torque-converting transmission.

As a result, attention was directed to the possibilities of a two-element transmitter on the Föttinger principle, but of one to one torque ratio

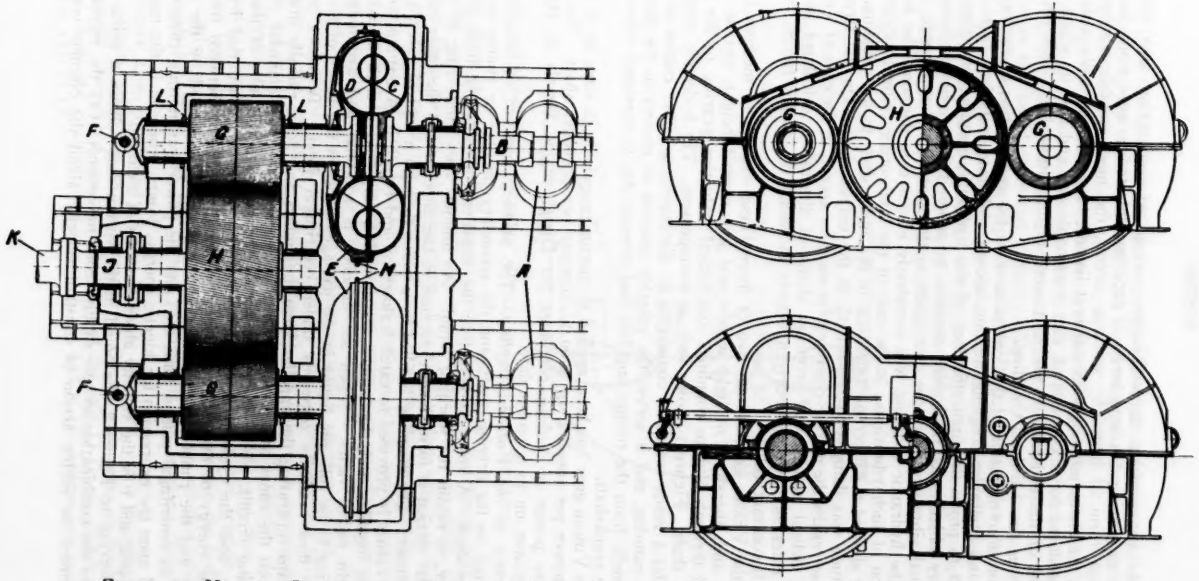

FIGURE 1.—VULCAN GEAR FOR THE PASSENGER MOTOR LINER "ST. LOUIS" OF THE HAMBURG-AMERIKA LINE.

intended for working in combination instead of competition with helical reduction gearing. A long series of experimental studies were made to find the best form of hydraulic working circuit, the most suitable number of vanes, the characteristics with different liquids, etc., and eventually a design was produced for a coupling of very reasonable size and cost in which efficiencies of the order of 98 per cent were readily attainable. This became known as the "Vulcan Coupling."

Prior to introduction of the Vulcan gear as the combination of mechanical gear and hydraulic coupling became known, elaborate tests had been conducted to prove the high efficiency of the hydraulic coupling and that the primary object was realizable; namely, to make it impossible to transmit torsional vibrations from the crankshaft to the gearing. Test records show that the hydraulic coupling was completely effective in damping the transmission of such vibrations; in the case of the sixth-order critical vibrations, which showed a dangerous amplitude in the input shaft to the coupling, the amplitude was damped to one-fiftieth in the output shaft. On the strength of these results the marine classification societies accept, in the case of Vulcan gears, helical gearing and propeller shafts of dimensions calculated by the same rules as for turbine-driven vessels in which the output torque has a smooth constant value.

In the Vulcan gear system two or more Diesel engines of medium high-speed and, therefore, of light weight and low overall height, can be connected through Vulcan couplings and helical reduction gearing to the propeller shaft. Figure 1 shows such an arrangement. The hydraulic coupling provides a ready means of connecting or disconnecting either engine at will while running, and it serves the valuable purpose of protecting the gearing absolutely from the torque impulses and stresses due to torsional vibrations of the crankshaft.

The Vulcan gear has been applied to marine propulsion in sizes up to 6750 horsepower per coupling or 2700 horsepower per propeller shaft. The aggregate power of Vulcan couplings for Diesel engine vessels is 443,250 horsepower up to a recent date.

Theory of the Vulcan Coupling.—The primary member or impeller is mounted on the driving shaft, and the secondary member or runner on the driven shaft. A cover is bolted to the impeller enclosing the back of the runner, to retain the working liquid. An annular core guide ring of semicircular section is formed with one-half in the impeller and the other half in the runner. The coupling is filled by a pump through a control valve, and a fixed casing is provided to catch leakage oil and return it to a sump tank.

The impeller and the runner are provided respectively with 28 and 26 straight radial vanes. When the impeller is rotating, and assuming the coupling to be full, the liquid in the passages between the vanes of the impeller flows radially outwards under centrifugal force. It passes across the gaps separating the impeller from the runner and flows radially inwards between the vanes of the runner until it reaches the inner diameter of the working circuit, where it returns across the gap into the inlet of the impeller, and the cycle is repeated. Since the power is transmitted by the kinetic energy of the rotating vortex of liquid circulating between the impeller and the runner, it follows that the design is governed by the same laws as centrifugal pumps, fans, etc., and the power-transmitting capacity is based upon the expression for kinetic energy, $\frac{1}{2}mv^2$, where m is the mass of the liquid and v is the velocity about the axis of rotation.

Referring to Figure 2, the liquid in the vortex ring follows a spiral path due to the combination of two simultaneous movements: (*a*) the circulation movement or vortex action of constant velocity about the circular core or

axis C C of the working circuit, and (*b*) the rotational movement due to the rotation of the liquid as a body with the driving and driven halves about the exis O O of the coupling.

Whereas the transmission of power is dependent upon the existence of the circulatory or vortex movement, the actual torque on the runner, for example, is created by the loss of rotational velocity and the release of kinetic energy as the liquid is forced radially inwards between the vanes of the runner. Conversely, on the impeller side, the liquid is flowing radially outwards between the vanes, and its rotational velocity about the axis O is increasing, thus taking up kinetic energy and imposing a corresponding load on the driving motor. Many wrong analogies have been presented to explain the working principle of the fluid coupling.

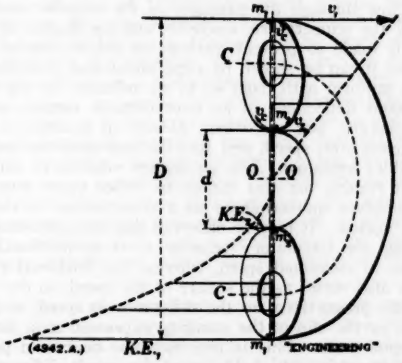

FIGURE 2.—THEORY OF ROTATING VORTEX OF LIQUID.

Thus, there is one idea that the circulation velocity is extremely high and hence the liquid becomes possessed of a wonderful resistance to shear, which somehow locks the two halves together. The familiar but wrong analogy is presented of the resistance to shear of a jet of water of high velocity when struck by a bar of iron. Another idea is that the liquid somehow acquires a high density or becomes extremely viscous under high centrifugal pressure, so that a positive drive is achieved. In actual fact, the drive is simply analogous to a centrifugal pump in conjunction with a reaction turbine.

The liquid gathers rotational velocity in its radially outward flow between the vanes of the impeller (or pump element) and imposes a corresponding load on the driving motor. Similarly, it loses velocity in its radially inward flow between the vanes of the runner (or turbine element) and thus gives up kinetic energy which drives the output shaft. The fluid friction loss resulting from the vortex or circulatory flow is small, because the body of liquid rotates with the coupling as a whole and the relative velocity between the circulatory liquid and the curved passages of the working circuit is small.

In Figure 2, m_1 equals unit mass of liquid at the outer profile diameter D; v_1, its velocity about the axis of rotation O O; m_2, unit mass of liquid at inner profile diameter d; v_2, its velocity about the axis O O; v_c, circulation velocity about the circular core or axis C C; and n, revolutions per minute about the axis O O.

During each half-cycle of circulatory flow the mass m_1 is considered to move to a new position m_2, and the kinetic energy at the respective positions is indicated by the lines K E_1 and K E_2. The change in kinetic energy is given by $\frac{1}{2}\, m(v_1^2 - v_2^2)$. Hence the power transmitted by a coupling of given diameter and speed is proportional to the mass or density of the liquid (corrected in practice for the degree of aeration). It is also proportional to the number of times this cycle is repeated in unit time, *i.e.*, the power transmitted is proportional to the circulation velocity v_c.

The velocity v_c is created by the difference in the centrifugal pressure of the column of liquid rotating with the impeller extending from d to D, and the counter centrifugal pressure of the corresponding column rotating with the runner at a speed which is slightly less owing to the slip. The circulation velocity v_c is affected by the frictional resistance encountered in its circulatory flow through the passages of the impeller and runner, but is also affected by the temperature, viscosity, and the degree of aeration.

No attempt is made to present methods of calculation of the circulation velocity v_c, since it can be shown by experiment that the liquid does not, in fact, follow the uniform path from m_1 to m_2 indicated by the diagram, but is subject to tortuous deflections of an unpredictable nature, which vary with the load, temperature, and viscosity. Hence, in practice, a constant K is established for each size, speed, and specific load upon the coupling to cover the many variables indicated. The equivalent velocity v^c can be readily deduced from test results, but this cannot be relied upon, since the values v_1 and v_2 are themselves indeterminate as a consequence of the tortuous path followed by the vortex. It may be observed that the difference in centrifugal pressure creating the circulatory velocity v_c is proportional to the square of the difference in rotational speed, whereas the frictional resistance to the circulatory flow also varies as the square of the speed, so that the velocity v_c should be directly proportional to the difference in speed of the driving and driven elements, or the slip of the coupling expressed as a percentage of the speed of the input shaft. This is true to some degree in practice, but the straight-line relationship only holds good for small differences. Hence it follows that the power transmitted by a coupling of given diameter and speed is proportional (between limits) to the percentage slip.

Having thus dealt with the factors m and v_c, it remains to consider the velocities v_1 and v_2, which are determined by the diameter D and rotational speed n, *i.e.*, to consider the influence of the dimensions and speed of the coupling respectively upon its power-transmitting capacity. Diameter has, of course, the greatest influence, since it determines the volume or mass of the liquid, as well as its changes in velocity. Speed affects only the changes in velocity, but is, nevertheless, of great importance. Assuming the diameter of the coupling to be fixed and that it is working at a constant percentage value of slip, the velocities v_1 and v_2 are proportional to the rotational speed n. Therefore, the kinetic energy given up by unit mass of liquid in moving from m_1 to m_2 will vary as n^2. Since the percentage slip is assumed to be constant, the number of cycles of circulatory movement in unit time will be proportional to n. Hence the rate of doing work or power transmitted is proportional to $n^2 \times n = n^3$.

The force required to accelerate the mass m from velocity v_1 to v_2 is proportional to the acceleration, and thus varies directly with the speed n. In addition, however, the number of cycles of circulatory movement in unit time (for the assumed constant percentage of slip) is proportional to n. Hence the force driving the runner and reacting on the impeller, or, in other words, the torque transmitted is proportional to $n \times n = n^2$.

Coming now to the influence of diameter, and assuming the speed of revolution and the percentage slip to be constant, it follows that the velocities v_1 and v_2 are proportional to D. Hence the kinetic energy given up by unit mass in moving from m_1 to m_2 will be proportional to D^2. In addition, the mass of the liquid varies with the volume of the working circuit, which is proportional to D^3. Thus the horsepower transmitted for a given rotational speed and percentage of slip is proportional to $D^2 \times D^3 = D^5$.

As already noted, the horsepower transmitted by a full coupling of given size and speed is proportional (within limits) to the percentage slip. In other words, the torque transmitted (for a constant speed of rotation) increases with the slip, reaching a maximum value when the driven shaft stalls, *i.e.*, when the slip is 100 per cent. The term "slip" is misleading to many engineers, since for a given degree of filling it only refers to loss of revolutions (as in the case of alternating-current electric motors), but not to loss of torque—in fact, the reverse. This confusion results, no doubt, from the significance of slip in a friction clutch, where the transmitting capacity falls off as the slip increases, so that slip is then synonymous with loss of both speed and torque. Nevertheless, since the term "slip" has become everyday practice in connection with fluid transmission, it is used by the author for lack of a better expression. Surprise and incredulity are still so frequently expressed regarding the small percentage of slip which is attainable with hydraulic couplings in everyday practice that it is opportune to say that there is not the slightest difficulty in designing a coupling to transmit any desired horsepower with a slip of 1 per cent or 2 per cent or even less. Generally speaking, it is not economical to design for less than 1½ per cent slip, while in many cases 3 per cent to 4 per cent slip is justifiable. The choice depends, of course, on the first cost, load factor, and running cost.

Figure 3 shows the present standard type of scoop-tube coupling in which the gravity tank, interconnecting piping, and ejector are eliminated, the oil being contained in a reservoir which supports the scoop-tube housing and is situated immediately beneath the manifold. Instead of an external control circuit, the manifold is provided with internal ports through which the oil delivered by the scoop tube is circulated and returned into the working circuit. The speed is adjusted by varying the quantity in the working circuit by means of a gear pump mounted on the reservoir, this pump being normally stationary. In order to increase the speed, the pump is rotated clockwise and draws oil through the suction valve in the reservoir, discharging it into the manifold, where it joins the stream of oil circulated by the scoop tube into the working circuit. Conversely, to reduce the speed, the pump is operated in the anti-clockwise direction and withdraws oil from the manifold under the pressure created by the scoop tube, discharging it through a spring-loaded foot valve into the reservoir chamber. Thus the transfer of oil from the reservoir to the working circuit, and *vice versa*, is immediate, and is proportional to the displacement of the pump. Two flanges are arranged on the manifold for connecting the flow and return pipes of an oil cooler in cases where cooling supplementary to the natural heat dissipation of the rotating casing is required. The scoop-tube pressure is sufficient to circulate the oil through the cooler without the aid of an auxiliary pump. On small couplings, hand operation of the pump is arranged, but the standard practice is to use an electric servo-motor with a reversing switch at a convenient remote-control point. The pump-controlled coupling eliminates any time lag due to interconnecting piping and overcomes any back-pressure effects due to the effect of variations in load upon the rotating vortex ring of liquid. Consequently, in its evolution, the design of the scoop-tube coupling has departed from the bad principle of regulating the rate of flow into the

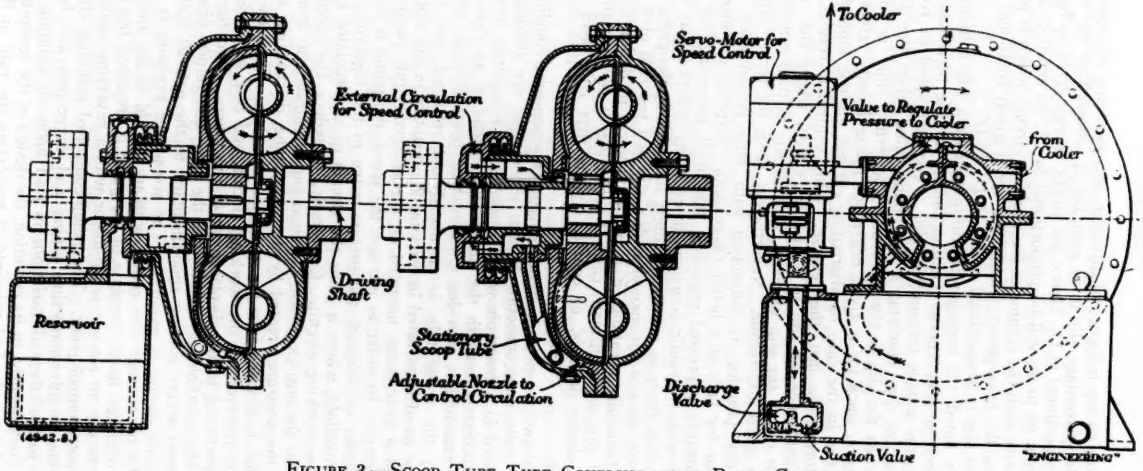

FIGURE 3.—SCOOP TUBE TYPE COUPLING WITH PUMP CONTROL.

coupling, with its many variables, and is based upon direct transfer of liquid to and from the working circuit.

As has been mentioned in the section on the theory of the hydraulic coupling, the circulatory movement of the vortex of liquid follows a tortuous path which is difficult to predict, although as the result of experimental study extending over many years, it has become possible to form a good idea of this path under particular conditions of filling and slip. During some early experimental tests at the Metropolitan-Vickers works on a 1000 horsepower coupling running at 1000 R.P.M. on a flywheel generator set, violent torque surges occurred when the flywheel was being accelerated between 250 R.P.M. and 400 R.P.M., with the consequence that the circuit breaker was repeatedly tripped. The flywheel in question had a very high inertia and a total kinetic energy of 23,000,000 foot-pounds. The flywheel would be accelerated at about 50 per cent of full-load torque, when the load on the motor would suddenly drop towards the light-load value. Then after a second or two, some violent internal convulsion would take place without warning, causing the load on the driving motor to increase instantly by ten or twenty times, thus tripping the circuit breaker. As the result of a long series of experimental trials, the cause of the vortex instability has been established, and an improved working circuit has been evolved which is free from such instability. In the case of constant-torque drives, where the horsepower varies substantially in proportion to the speed of the driven shaft, the hydraulic coupling is required to work through the unstable period, and the "surgeless circuit" must, therefore, be used to eliminate all instability.

The essential feature of the "surgeless circuit" is the introduction of a suitable discontinuity in the boundary of the working circuit, either in the curved shell or the sides of the vanes, against which the circulating liquid reacts in giving up kinetic energy. A convenient discontinuity is formed by a star-shaped baffle fixed in the neighborhood of the inner profile diameter and projecting into the working circuit to a sufficient degree to break up instantly a high-velocity vortex surge. A star baffle of this kind is sufficient to reduce the surges to a tolerable figure. To gain complete freedom from surges a circular baffle is fixed near the inner profile diameter, and the small increase in slip due to this obstruction can be eliminated by an increase of 5 per cent or so in the diameter. The same results can be achieved by curving the vanes of the impeller or runner, or both. Such interferences with the working circuit are attended by an increase in slip, which is of a negligible order in the case of a suitably designed baffle. In the case of inclined vane constructions, however, the increase in slip is greater than with a baffle, and unless the angle of inclination is correct, the slip can easily reach an intolerable value.

Hydraulic Thrusts.—In the marine type of Vulcan coupling, working in combination with helical gearing, the standard practice is to balance the hydraulic thrust partially against the thrust of the helical gear, and to provide thrust collars on the driving and driven shafts to deal with any unbalanced load. Under normal conditions, with the coupling full, there is a hydraulic thrust in the outward direction, *i.e.*, tending to separate the impeller and the runner. This thrust is considered to be due to the centrifugal pressure set up by the vortex action, counterbalanced by the pressure on the back of the vaned wheel (usually the runner) which is enclosed by the inner casing. When a coupling is operating in a partly-filled condition, however, the vortex ring alters in shape and the thrust load reverses under certain load conditions, tending to draw the impeller and runner together. In the early designs of scoop-tube coupling based on the marine coupling, provision was made for outward thrust only, a plain bronze pad or distance

piece being used to space the impeller and runner, with the result that wear was quickly experienced. Fortunately space permitted the substitution of a ball-thrust bearing of sufficient capacity, which overcame the difficulty. For certain types of drives, a unidirectional ball thrust meets all requirements, but in cases where the thrust load is likely to reverse its direction, the practice is to use a duplex thrust bearing.

Where the specific load and speed of the coupling are high and the thrust load is likely to be of some magnitude, it is possible to reduce it by coring or cutting large ports through the shell forming the back of the runner (or the vaned wheel enclosed by the casing). Such ports must be of a suitable size and location to be of any use, as shown by Figure 4, and as they have an influence on the slip, their use must be judged by circumstances.

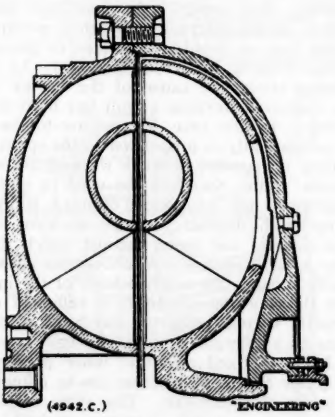

FIGURE 4.—THRUST REDUCING PORTS IN DRIVEN MEMBER.

They have not, in consequence, become standard practice for all sizes of couplings. For industrial applications in general, the use of thrust bearings within the coupling is much to be preferred to external thrust bearings, particularly as the work done on the single-thrust bearing is simply proportional to the load multiplied by the slip revolutions or the difference in speed, whereas with external thrust bearings the work done on each of the two bearings is the product of the load and the speed. A coupling provided with an internal thrust bearing must be considered equivalent to a solid coupling in an axial sense, although it provides lateral and torsional flexibility. Hence, to allow for expansion effects, the standard practice is to locate one shaft and allow the other to float. The motor shaft is usually the more convenient to locate by a deep-groove ball bearing, which is not, of course, required to take any hydraulic thrust. If the driven shaft is located by thrust collars, the motor must have freedom to float axially.—" Engineering," May 3, 1935.

BOOK REVIEW.

ELEMENTS OF DIESEL ENGINEERING, By Orville Adams. The Norman W. Henley Publishing Co., 2 West 45th St., New York. Price $4.00.

This volume, intended primarily to combine into a single volume a text book and a guide for specializing in operation and maintenance, covers in rather tabloid form the entire field of American Stationary, Marine, Locomotive and Automotive Diesel Engineering. Almost every known type of standard Diesel engine and Diesel engine accessory in this category is described and illustrated by up-to-date cuts. The book smacks of modern trends in Diesel development. There is no attempt to exhaust any of the subjects treated, the information given being the essential working principles of construction, operation and service of Diesel engines well suited to students' use.

Chapters II, III and IV, covering Fuel Injection Nozzles, Fuel Injection Pumps and Combustion Chambers, are especially well written. There are practical variables which often enter into every thermodynamic equation related to combustion phenomena which can not readily be put on a paper, but a careful analysis of the aforementioned chapters should give a student a good understanding of the essentials of proper combustion and its function in the Diesel cycle. The author very wisely gave emphasis to these factors by placing them in the first part of the book.

The work reflects in every way the wide experience of the author. It should be particularly valuable to the young Diesel engineer getting his first practical experience. It merits the careful investigation of the skilled designer and seasoned technician.

L. F. S.

ASSOCIATION NOTES.

ANNUAL BANQUET.

The Annual Banquet of the Society was held this year on April 3, 1936, at the Willard Hotel in Washington. More than eight hundred members and guests attended. Admiral William H. Standley, U. S. Navy, Chief of Naval Operations, and President of the Society, presided and Rear Admiral S. S. Robison, U. S. Navy (Retired), acted as Toastmaster.

Following the President's address by Admiral Standley, Rear Admiral Adolphus Andrews, U. S. Navy, Chief of the Bureau of Navigation, spoke on "Engineering Personnel in the Navy." Dr. W. R. Whitney, Vice President in Charge of Research, of the General Electric Company, followed Admiral Andrews and chose as his subject "Simple Nerve." Rear Admiral H. G. Bowen, U. S. Navy, Chief of the Bureau of Engineering, closed the speaking with a brief talk, entitled "Remarks on Engineering." It is regretted that no copies of the latter three excellent addresses are available for publication.

Admiral Standley said:

By a fortunate combination of circumstances I am privileged again to be with you in this annual gathering which has come to mean a great deal to all of us, and especially to me, your President. I returned to the United States only yesterday from the Second London Naval Conference.

When I left Washington I expected either to be back earlier or to stay longer. In the former event, I would be home in ample time to arrange for this party; in the latter event, I could not possibly come. As the time for the dinner approached and there seemed to be no possibility of my getting back, a decision was made eliminating me from the program. However, when it became certain that we would leave on the steamer of March 26th, arriving in the States on April 2nd, your secretary sent me a cable saying they

would expect me at the dinner and I could make the President's address, which might only take the nature of impromptu remarks, if I chose—so that is my status here tonight.

As for my text, I choose to tell you briefly the results of our efforts in London. The Conference was mandatory, both under the London Treaty and as a result of Japan's notice of termination of the Washington Treaty, and was called in an endeavor to reach an agreement which would take the place of both those treaties.

It would perhaps be interesting to sketch the practical working of the Conference. The only plenary sessions of the Conference were an opening session to welcome the Delegations and to allow each nation to make an initial statement of position and a closing session to sign the Treaty. At the opening session it was evident that the only point on which there was general agreement was a desire for a reduction in the cost of armaments.

The first meeting was held in the Locarno Room at the Foreign Office, London, on Monday, 9 December 1935. Mr. Stanley Baldwin presided and delegates from the British Empire, Japan, France, Italy and the United States were present. Sir Samuel Hoare and Lord Monsell, respectively, were unanimously elected President and Vice President of the Conference.

Japan advocated the abolition of all capital ships, aircraft carriers and 8-inch cruisers, but desired for herself an increase in submarine strength. In effect, this proposal was a demand for parity with the navies of Great Britain and the United States based on "global" tonnage, with freedom to apportion such tonnage as might best suit her needs.

Summarizing the positions outlined by each Nation, Britain wanted smaller and therefore cheaper ships and more of them; Japan, France and Italy would possibly agree to an all-around reduction in size. The United States was willing to reduce numbers but not the lowering of tonnages for individual ships, believing that our problem, being without adequate bases, was to have vessels with sufficient radii of action and sea keeping qualities to meet our peculiar needs. France and Japan would give no consideration to abolition of submarines. Italy's demand was for a Navy at least equal to that of France.

Japan's proposal for parity with Great Britain and the United States was not well received.

At this plenary session a First Committee was also appointed, composed of Delegates and Technical Advisers, which was in the nature of a Steering Committee. Each Delegation repeated in greater detail before this committee its position and arguments in support thereof. The meetings were very formal and every speech had to be translated from French to English and at times from Japanese to English to French, which naturally resulted in a twenty-minute speech consuming an hour's time. The First Committee was too unwieldy a body to handle details and various subcommittees were appointed. To thresh out technical questions, a Technical Sub-committee in turn appointed certain members to report on detailed questions. The real work of reaching an agreement was, however, done through bilateral conversations outside of any committee. When as a result of these conversations a sufficient measure of agreement had been reached to make a meeting of a committee useful, the committee was called and threshed out remaining minor points of difference. Thus was the treaty built up.

You know, of course, that the Japanese left the Conference because their thesis of a common upper limit was not acceptable to any of the other Powers. When this happened or became a certainty, counsel was taken among the remaining Delegates as to whether to give up and go home or whether to continue in session in an endeavor to salvage something from the wreck of the Washington and London Treaties. Home governments were consulted and possible accomplishments considered resulting in the decision to continue. Quantitative limitation went out with the Japanese, but this did not become definitely established until the Conference again met and the proposals of the various Delegations were again considered. I will not bore you with the details of our deliberations. Suffice it to say that the difficulties in reaching agreements on any point were enormously increased by the fact that only about half of the important Naval Powers were represented. We first agreed upon definitions defining various naval terms including all existing types and categories of naval vessels. This was the starting point and common meeting ground of understanding. We next

agreed upon a plan of announcing in advance our yearly building program and of adhering to this program except under emergency conditions and in the manner provided in the treaty itself.

We next agreed to limit our building to existing and agreed types, again except under emergency conditions and as provided in the treaty itself. This is the so-called qualitative limitation—sizes of ships and guns being limited but no limit being placed upon the number of the types which could be built.

The fourth part of the treaty is the so-called "safeguarding clause" section. This section is necessary because, as previously stated, so many of the Naval Powers were not parties to the treaty and were in no way affected by the treaty, and no signatory power would agree without such an escape clause to tie her own hands in the face of a possible threat to her national security by the naval building of a non-signatory Power. There was also the escape in case of war—and the escape on account of circumstances which could not be foreseen. This section was the most complicated and the most difficult to write. It necessarily had to cover all contingencies for all the signatory Powers. Then it had to provide for the steps to be taken in each case and the manner in which it then was to be followed. The agreements finally reached are a tribute to the patience, tact, and sound judgment of the Technical Committee which handled this matter.

The final section provides for consultation and exchange of information, as to whether or not another Conference should be called and as to the possibility of a reduction in the size of battleships to be built after such consultation; this consultation to take place in 1940. The Treaty expires December 31, 1942.

The actual work of preparing this Treaty could easily have been completed in a month's time, but one thing after another interfered with our work. First, the uncertainty as to just what the Japanese were going to do and the definite determination of the details of this situation occupied our time up to beginning of the holiday period. This period lasted from December 21st to the 7th of January. Then there was considerable delay in preparing the way for the Japanese exodus and in deciding what to do then. The week of official and intensive mourning on account of the King's death caused another delay. Then political complications for the

European Powers began to develop political complications in which we had no part or vital interest except that it delayed the work of the Conference—complications which involved England, France, Italy, Germany, Russia and Japan. Nearly a month's time was lost through this cause and it is curiously interesting that this matter was eased by the action of Germany in occupying the Rhineland. However, the Italian Delegation announced that on account of various provisions in the treaty and for economic reasons they would not be able to sign now, but they continued on with the Conference and the members of their Delegation gave valuable assistance in solving the various problems.

Thus we brought home a Treaty which in many respects falls far short of what we had hoped. For those who expected radical reduction in armaments it is a keen disappointment because the treaty not only visualizes but prepares the way for increase rather than reduction in naval armaments. On the other hand, in the face of obstacles which confronted us and in view of the instability in world affairs, it seems rather a miracle that we could reach any agreement whatever. Furthermore, if we can forget our disappointment at our failure to obtain real reduction through quantitative limitation we can console ourselves with the feeling that the information contained in the announced programs and the qualitative limitations provided will allay suspicion and thus tend to avoid a naval race which after all is the reason for the great cost in armaments. If nations can be persuaded to agree as to their actual needs and to build accordingly and not against each other we will have a real limitation in armaments. We have also, by this Treaty, continued the structure for the limitation and reduction of naval armaments set up by the Washington Treaty, a structure in and through which a real reduction of armament may later be brought about. It seems to me that in spite of our disappointment, the Treaty which we have signed is well worth while.

Now in conclusion I want to refer to the plans and policies concerning our own navy plans with which you as naval engineers are vitally concerned and which as American citizens you should advance in every possible way.

Our present policy is to build our navy to treaty strength and to maintain it there, "treaty strength" being, in this instance, that

prescribed in the Washington and London Treaties. From plans based on this policy, we already have under construction all the tonnage required to bring the navy to actual treaty strength and the replacement of obsolete tonnage in destroyers and submarines will be completed in 1942. Provision for the replacement of obsolete battleships was held in abeyance pending the result of the Conference of which I have just spoken but our plans contemplate their inclusion in our replacement program.

Of course, with the expiration of the Washington and London Treaties at the end of this year, treaty strength as a measure of naval power will no longer hold, and the proposed treaty contains no similar measure. It must then be realized that, with the expiration of the Washington and London Treaties, any naval power will be free to engage in unlimited building both as to numbers and types. The proposed treaty will provide some limitation as to the types of ships which can be built but, within this scope, any power can build ships in unlimited numbers. However, such a procedure, if adopted, must result in the much dreaded naval race, and it is believed that no power can afford to start such a race. We, ourselves, have stated that we will adhere to our previously announced policy and plans, unless the building of some other power or powers forces us to depart therefrom.

However, I cannot escape the conviction that an increase in naval armaments is contemplated by most of the naval powers. We know England has plans, and has already started an increase in her cruiser strength. With the existing world conditions, we must retain our relative naval strength and the probabilities are that we may have to reconsider our previously planned 1942 building program. With the Washington and London Treaties gone it becomes more important than ever to have at all times a balanced and continuous building program which will permit us to profit by our errors and successes in the complicated design of modern vessels of war and to parallel at all times, to the greatest possible extent, technical development in commercial and industrial fields. Other powers have such a program and unless we also do so we shall soon be hopelessly outclassed in modern and up-to-date ships. Complete shipbuilding holidays of extended duration seriously imperil our national security.

If naval powers will keep other powers fully informed in the manner provided in the proposed treaty as to their intentions I feel that this will do much to prevent the so-called naval race.

I desire to assure you once more of my deep pleasure in being able to greet you tonight and to thank you for joining with us in a well-earned moment of relaxation which marks another milestone of achievement in naval engineering fields.

I am glad to be home.

CHANGES IN OFFICERS.

Due to detachment from duty in Washington, Commander C. S. Gillette, U.S.N., and Commander Lybrand P. Smith, U.S.N., have resigned as Secretary-Treasurer and member of the Council, respectively. Commander Gillette's resignation will be effective 1 June, 1936. He will be succeeded by Commander Roger Ward Paine, U.S.N. Commander Smith will be succeeded by Lieut. Commander R. W. Bates, U.S.N.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

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Dow, Jennings B., Lieutenant, U. S. Navy.

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Jeans, Howard S., Commander, U. S. Navy.

Lamons, Ernest W., Lieutenant, U. S. Navy.

Loeser, Arthur E., Lieutenant, U. S. Navy.

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Murdaugh, Albert C., Lieutenant, U. S. Navy.

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